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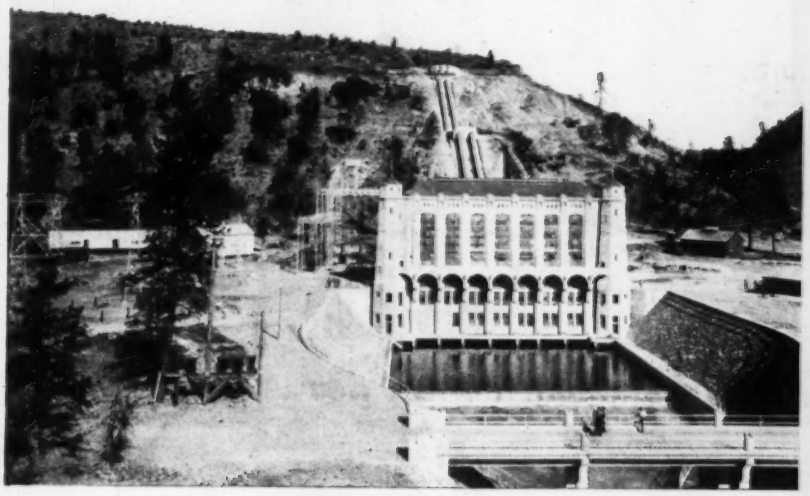
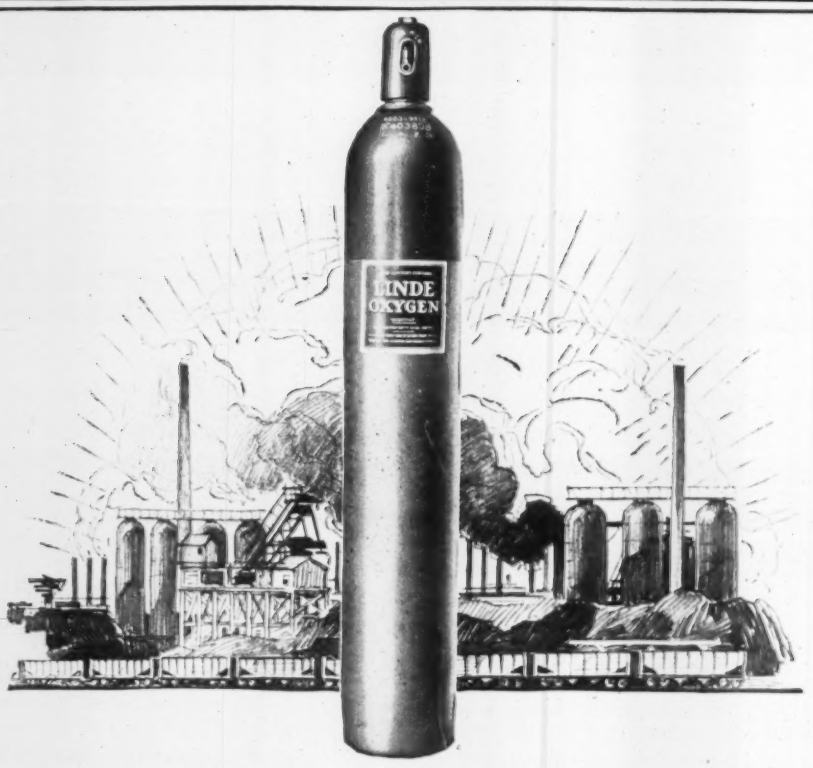


EXTERIOR OF PIT RIVER POWER PLANT NO. 1

See article in this issue on The Electrical and Industrial West

AUGUST 1923

THE MONTHLY JOURNAL PUBLISHED BY THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS



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Volume 45 August, 1923 Number 8

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ROBERT SIBLEY

H. G. ACRES

MYRON A. LEE

PETER PAYNE DEAN

F. L. KALLAM

Contributors to This Issue

Robert Sibley was born at Round Mountain, Ala., in March, 1881. He was graduated from the University of California in 1903 with the degree of B.Sc., and during the next four years was professor of mechanical and electrical engineering and head of the School of Engineering at the University of Montana. In 1907 he opened an office in Missoula, Mont., as consulting engineer in hydroelectric investigations in the Northwest. He returned to California in 1911 as associate professor of mechanical engineering at his Alma Mater, becoming professor the following year and remaining in that capacity until 1915. He was given a doctorate degree in electrical engineering by that institution last year.

During the next year Dr. Sibley travelled in Japan, China, the Philippines and various industrial centers in eastern and southern parts of the United States. July, 1916, found him back in California once more, as editor of the *Journal of Electricity*, and in 1920 he was appointed president of the McGraw-Hill Company of California. He resigned these positions early this year to become executive manager of the University of California Alumni Association, but still serves the McGraw-Hill Co. as Pacific Coast Consultant.

Dr. Sibley is a vice-president of the A.S.M.E. and a fellow of the A.I.E.E. He is not averse to boosting California's climate on occasion, but his voice is heard more frequently on the subjects of fuel oil for steam-electric generation, and the hydroelectric developments of the Pacific Coast, on both of which he is an authority.

* * * * *

Charles E. Brooks, whose paper on the Recent Development of the Motor Coach is included in this issue, was born at Constantinople in July, 1886. He received his B.Sc. from McGill University in 1908, and soon afterward entered the employ of the Grand Trunk Pacific Railway. He served this railway in various capacities, finally as superintendent of motive power, having supervision over all work in connection with maintenance of locomotives, cars, and shops. When the Canadian lines were recently merged into the Canadian National Railways, Mr. Brooks was appointed chief of motive power of the new organization. The preparation of the paper in this issue at a time when he was assuming the manifold duties of a new position, as well as transferring his headquarters from Toronto to Montreal, is in itself no small achievement.

* * * * *

H. G. Acres writes on the subject of Modern Hydraulic Turbines of Large Capacity. Mr. Acres was born in Paris, Ontario, in 1880. He was graduated by the Faculty of Applied Science, University of Toronto, with the class of 1903. After a short term in the engine room of Canadian Pacific Railway liners between Montreal and Liverpool, he entered the employ of the Canadian Niagara Power Company.

In the Spring of 1905 Mr. Acres went to Arizona as assistant to the chief engineer of the Arizona Copper Company, working on railroad and mill construction. In the Fall of that year he returned to Canada as hydraulic engineer of the Hydro-Electric Power Commission of Ontario, which position he still holds.

Mr. Acres is a member of the Engineering Institute of Canada, the A.S.C.E., and the A.I.E.E. We cannot resist remarking that for a young man Mr. Acres has covered a good deal of ground.

* * * * *

Myron A. Lee was born in Auburn, N.Y., in March, 1887, and was educated in the public schools there and at Cornell University. After his graduation he took the engineering-apprentice course offered by the Western Electric Company and was then employed by that company as equipment engineer. He returned to Cornell, first teaching machine design and then industrial engineering. He took his M.M.E. degree during this time.

Professor Lee left Cornell to accept a position with the Thomas-More Aircraft Corporation, and was assistant superintendent in their plant during the war. He returned to college teaching after the armistice, and is now assistant professor and head of the Department of Industrial Engineering at Cornell.

* * * * *

Peter Payne Dean, who was responsible for the complete design and carrying out of the valve test conducted by the Public Service Electric Co. at their Essex station, Newark, N. J., was born in Staffordshire, England. He attended the Birmingham Technical School and the City and Guilds of London Institute. He served his apprenticeship with Vickers' Sons & Maxim, and then became assistant engineer with the Sturtevant Engineering Co., of London. In 1909 he came to the United States as sales engineer for the Diehl Manufacturing Co. of Elizabeth, N. J., and during his association with this company he completely developed an electrical dynamometer extensively used in the scientific testing of internal-combustion engines.

In 1914 Mr. Dean became associated with the Cutler-Hammer Manufacturing Co., which took over his patents on valve control. Since then he has devoted his time to the design and manufacture of valve-control systems and remote-control devices for central-station use.

* * * * *

F. L. Kallam, who has been awarded an A.S.M.E. prize for his paper on the Thermal Conductivity of Liquids, received his A.B. from Leland Stanford Jr. University in 1921, and his M.E. in 1922. He has been connected with several iron and fuel companies in California since graduation, and is now engaged in the natural-gas-gasoline industry with the Shell Company of California, as gas engineer.

Making America Independent in the Air

Fundamental research in the problems of flight must be the basis for adequate development of air service, either commercial or military. The National Advisory Committee for Aeronautics is in charge of an extremely valuable research program which will be treated in the leading article in the September issue of MECHANICAL ENGINEERING. Many interesting problems and their successful solutions will be described.

MECHANICAL ENGINEERING

Volume 45 August, 1923 No. 8

The Electrical and Industrial West

A Commercial and Industrial Empire Looks to the Engineer for Its Upbuilding by the Utilization of Its Vast Waterpower Resources

By ROBERT SIBLEY,[1] BERKELEY, CAL.

THE PURPOSE of this article is to give to the reader a conception of some of the unusual engineering feats that are being accomplished west of the Rocky Mountains and to point out some of the reasons why they have been attempted—and in a word to call to the attention of fellow-engineers the vision of the future that prevails among our engineering brothers in that great section of America lining the Pacific.

Due to the scarcity of coal and the necessity for irrigation combined with proper control of flood water and its economic application, the West stands preëminent in irrigation, in flood control, and in hydroelectric development as regards other sections of the world. Hence the engineer here is largely concerned with the various ramifications of water and the utilization of hydroelectric power down through the industries. Nature seems to have foreseen in its bounteous provision for our citizens west of the Rockies the crying need for the service of water. In the eleven states west of the Rockies, constituting but 8 per cent of the population, we find 70 per cent of the great undeveloped water powers of the nation. Already great strides have been made in the utilization of this great natural resource. West of the Rockies the average use of electrical power per capita is twice the per capita consumption of the nation as a whole, while in hydroelectric energy the citizen west of the Rockies uses six times the hydroelectric energy consumed by the average citizen of the nation, and in hydroelectric power still undeveloped he has a per capita possibility of twenty-two times that of the average citizen of the nation, even though the East has its great Niagara, Muscle Shoals, and other large power resources.

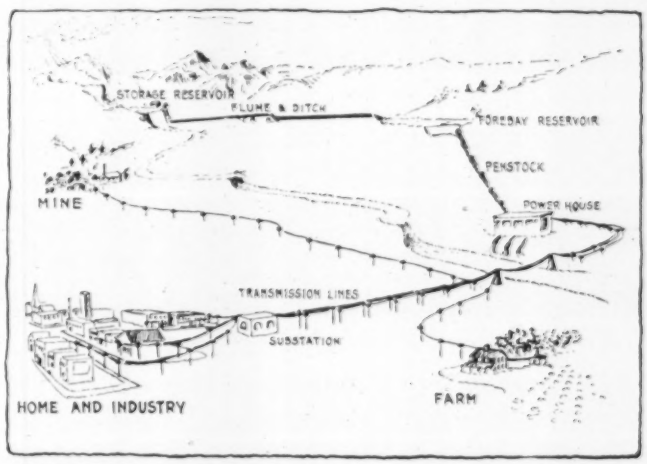


FIG. 1 SKETCH ILLUSTRATING A TYPICAL MODERN HYDROELECTRIC POWER SYSTEM ON THE PACIFIC COAST

The general conception of hydroelectric power as it has been developed to date in the West is depicted graphically in Fig. 1. There are no great Niagaras in this empire west of the Rockies, hence the engineer has found it necessary to create artificial waterfalls. His general method is to go up into the mountain gorges, build a dam to store the water, convey this water through a ditch or tunnel, often ten to fifteen miles around the mountain side, at a gradient less than that prevailing in the main run of the stream, and thus create an artificial waterfall of 1000, 1500, and often even greater than 2000 ft. vertical drop. Through giant pipes these waters are dropped against water wheels often of record proportions, thereby developing electrical power which in turn is transmitted to the farm, to the mine, to the home, and to the various industries in the great cities of the West.

Let us by pictures follow through some of these record-breaking installations. Fig. 2 is a view of Mt. Shasta in Northern California, the head of a vast drainage area for future power development. Fig. 3 shows some of the many volcanic springs existing to the east of this great drainage area which make possible a constant source of water for power development. The Pacific Gas and Electric Company has undertaken the development of some 600,000 hp. in this region, known as the Pit River development. This company, the largest of its type in the world, ties in with five other great power companies of California, in an interconnected network of approximately a million and a half horsepower, over an area greater than that of New York State, Pennsylvania, and all the New England states combined. Last year the total generated energy of this network was in excess of 4,300,000,000 kw-hr. Assuming the daily work of a human being as the equivalent of one kilowatt-hour, this means that the harnessing of California's mountain streams made possible last year the delivery of energy equivalent to a day's work of four billion, three hundred million people or two and a half times the entire population of the world today.

The view on the cover of the present issue shows the exterior of Pit River Power Plant No. 1. To the left will be seen the power lines which transmit hydroelectric energy at the record-breakage tension of 220,000 volts. This line is 202 miles in length and journeys southward to San Francisco and the industrial centers of central California.

Fig. 4 shows the interior of this power plant constituting the largest 60-cycle generator yet put into commission. The total output of each generator is 46,000 hp.

Fig. 5 is a close-up view of the 220,000-volt transmission line. In the building of this line the copper required for the conductors amounted to 10,000,000 lb., the largest amount ever placed in a single order in an engineering development of this nature.

Many interesting world records have been established in the construction of such great transmission lines as this. One of them is the well-known span at Carquinez Straits near San Francisco, where we find a single span of wire over 4600 ft. in length. Another is the submarine transmission of power from Berkeley, Cal., into San Francisco, through a cable beneath San Francisco Bay, a distance of 7 miles, at 11,000 volts.

In the southern portion of California the Southern California Edison Company is likewise breaking many world records in engineering achievement. The Big Creek Development which is being financed and engineered by this company, has as its goal the ultimate development of 1,400,000 hp. on the various tributaries of the San Joaquin river; particularly, however, on the tributary known as Big Creek. This company has the distinction of having in its interconnected system the successor to the old San Antonio

[1] Pac. Coast Consultant, McGraw-Hill Co., Exec. Mgr., Cal. Alumni Assoc., Stephens Union. Vice-Pres. A.S.M.E.

Abstract of an illustrated address given before the Pacific Coast Regional Meeting of THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS, Los Angeles, Cal., April 16 to 19, 1923.

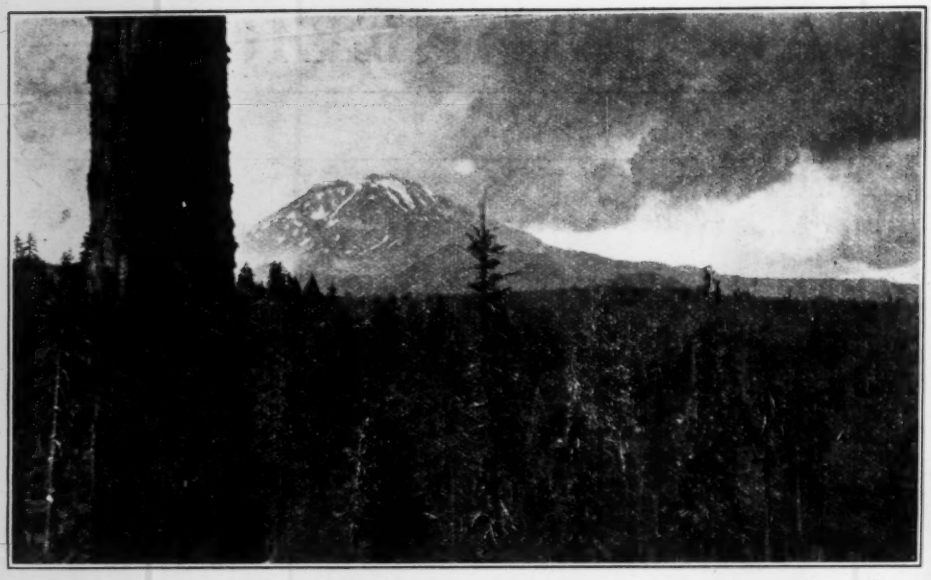


FIG. 2 MT. SHASTA, IN NORTHERN CALIFORNIA, THE HEAD OF A VAST DRAINAGE AREA FOR FUTURE POWER DEVELOPMENT

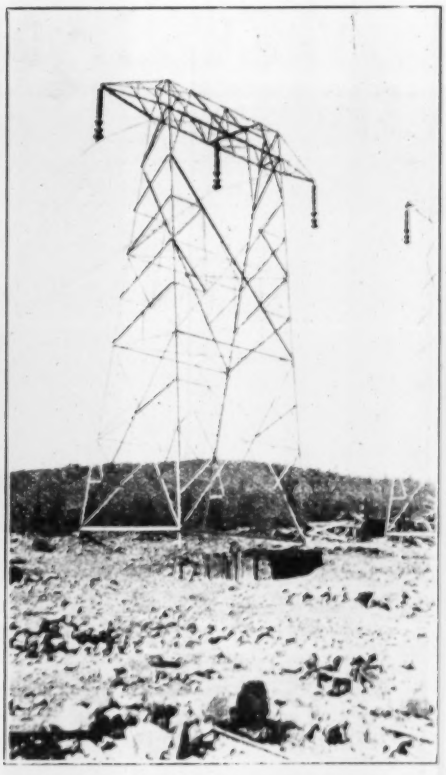


FIG. 5 (AT THE RIGHT) VIEW OF PIT RIVER 220,000-VOLT TRANSMISSION LINE

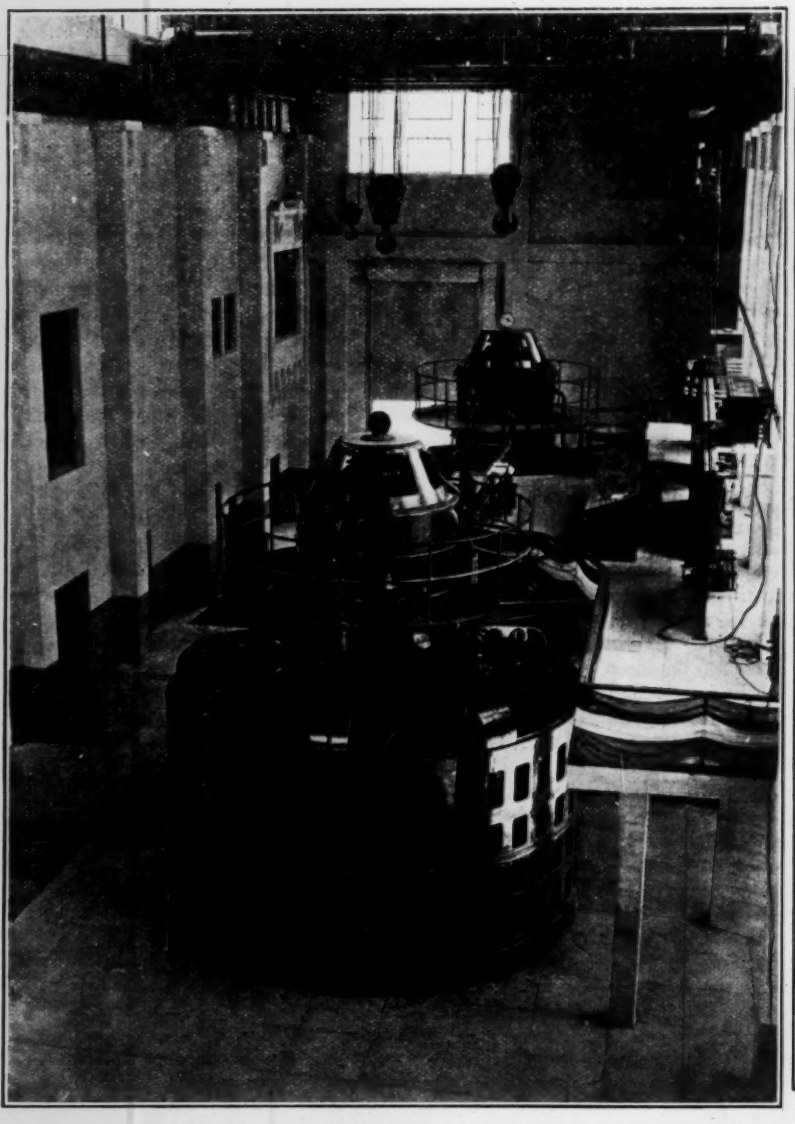


FIG. 4 INTERIOR OF PIT RIVER POWER PLANT NO. 1, SHOWING THE LARGEST 60-CYCLE GENERATORS YET BUILT. OUTPUT, 46,000 HP. EACH

FIG. 3 VOLCANIC SPRINGS OF THIS NATURE EXIST TO THE EAST OF THE MT. SHASTA AREA AND FORM A CONSTANT SOURCE OF WATER FOR POWER DEVELOPMENT

FIG. 6 EXTERIOR VIEW OF BIG CREEK POWER HOUSE NO. 1. 64,300 HP. INSTALLED; TOTAL HEAD, 2130 FT.

FIG. 7 ROAD CONSTRUCTION ALONG SAN JOAQUIN RIVER CANYON BETWEEN POWER HOUSES NOS. 8 AND 3 ON BIG CREEK PROJECT

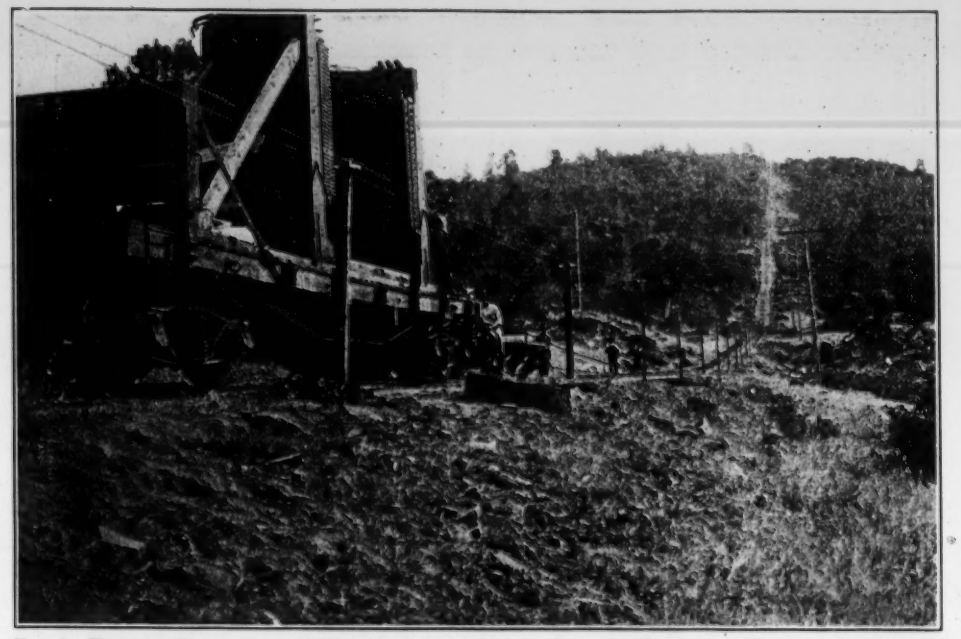


FIG. 8 HALF OF STATOR FOR 25,000-KW. GENERATOR AT BIG CREEK POWER HOUSE NO. 3 BEING LOWERED ON INCLINE TO POWER HOUSE SITE

FIG. 9 VIEW OF BIG CREEK POWER HOUSE NO. 8, IN WHICH MAY BE SEEN THE SURGE CHAMBER AND PENSTOCK

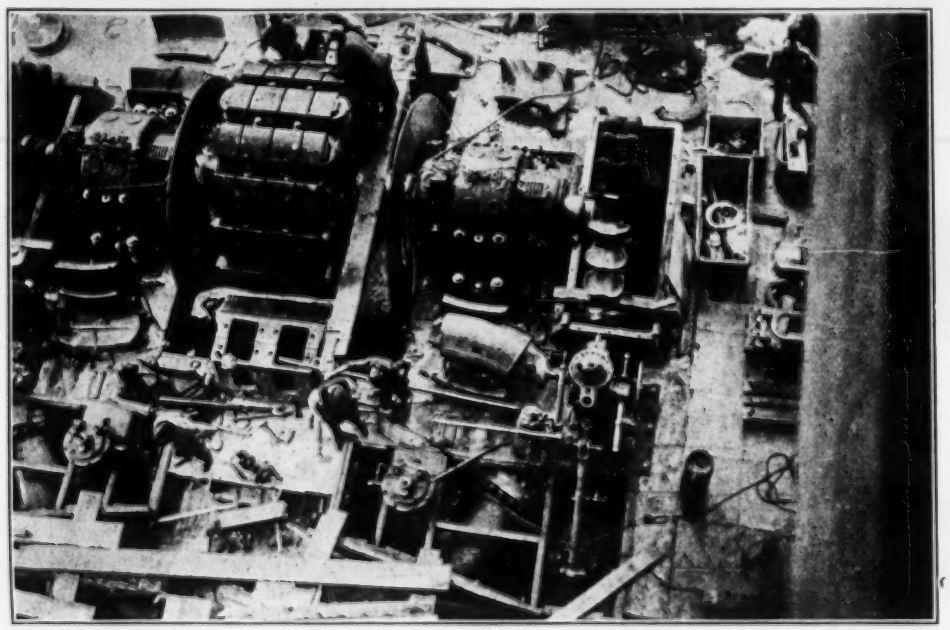


FIG. 10 LOOKING DOWN ON THE THIRD 16,000-KW. UNIT BEING INSTALLED AT BIG CREEK POWER HOUSE NO. 1, SHOWING GENERATOR IN MIDDLE AND AN OVERHUNG TANGENTIAL WATER-WHEEL AT EACH END OF THE SHAFT

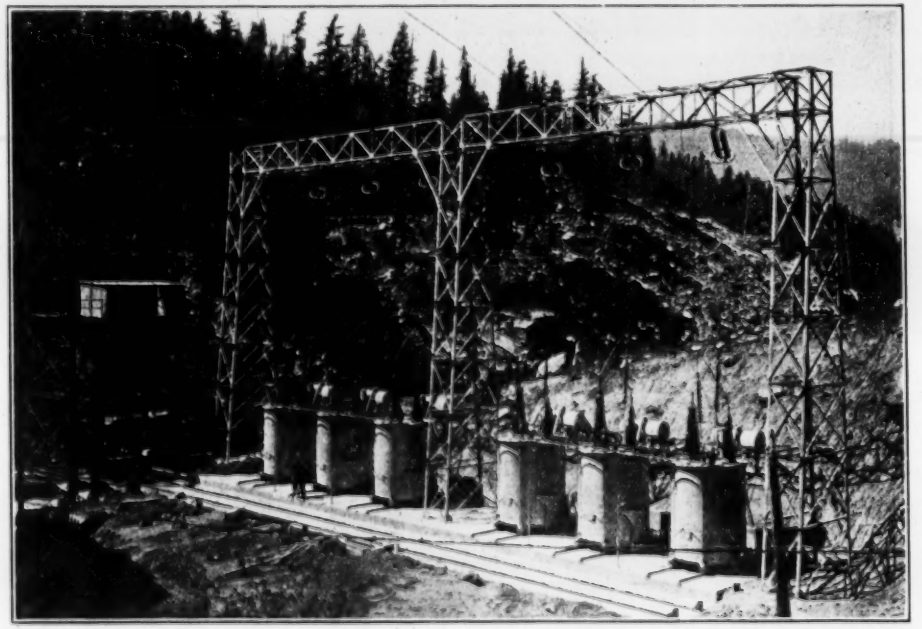


FIG. 11 TWO RECORD-BREAKING BANKS OF AUTOTRANSFORMERS AT BIG CREEK POWER HOUSE NO. 1, USED TO STEP UP VOLTAGE FROM 150,000 TO 220,000 VOLTS. EACH BANK IS RATED AT 52,500 KVA.

power plant, built in 1892, which in its day held the world record for hydroelectric transmission and was the first to transmit power commercially at a higher voltage than that at which it was generated. The length of the transmission line was 20 miles and the voltage of transmission 10,000. Today the giant lines of this same company are transmitting power from this Big Creek Development on the San Joaquin river into Los Angeles, a distance of 240 miles, at a voltage of 220,000.

Records of other California companies are also interesting. It must be remembered that California is a state of such proportions that if the northern boundary could be placed at the Woolworth Building in New York City the southern boundary would be somewhere near Jacksonville, Florida. The Southern Sierras Power Company, flanking the eastern slope of the Sierras in California, which delivers its energy to the deserts to the south in and about the Mexican border line, transmits its power at 87,000 and 55,000 volts over the record distance of 539 miles.

In the matter of sizes of turbine units, interesting records have been established. The two 30,000-hp., 1008-ft. head units in the Caribou plant of the Great Western Power Company in Northern California comprise the largest high-head impulse turbines thus far put into commission.

Fig. 6 shows an exterior view of Big Creek Power House No. 1, of 64,300 hp. installed capacity, which uses water under a total head of 2130 ft. and is the highest-head hydroelectric plant in America. It will be interesting and instructive to follow in some detail the difficult and unusual processes necessary in the engineering construction and installation of this great record-breaking accomplishment.

Fig. 7 shows the difficult road construction necessary in the San Joaquin River canyon between power houses Nos. 8 and 3 of the Big Creek project. Miles upon miles of such construction are often necessary before installation work proper can be undertaken in western hydroelectric power development.

Fig. 8 shows half of the stator for one of the generators in Big Creek Power House No. 3. This stator is for a generator of 25,000 kw. capacity and is shown in the process of being lowered on an incline down to the power house.

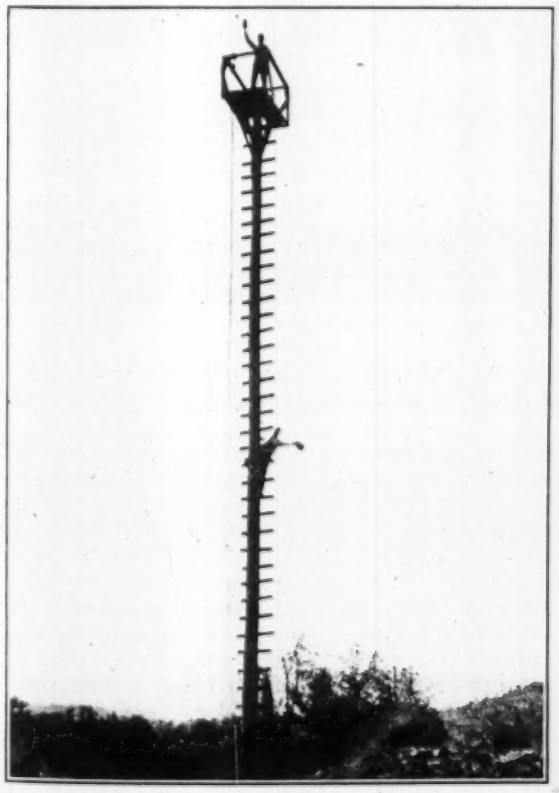


FIG. 12 TEMPORARY LOOKOUT FROM WHICH ENGINEERS TAKE PHOTOGRAPHS DAILY SHOWING PROGRESS OF WORK

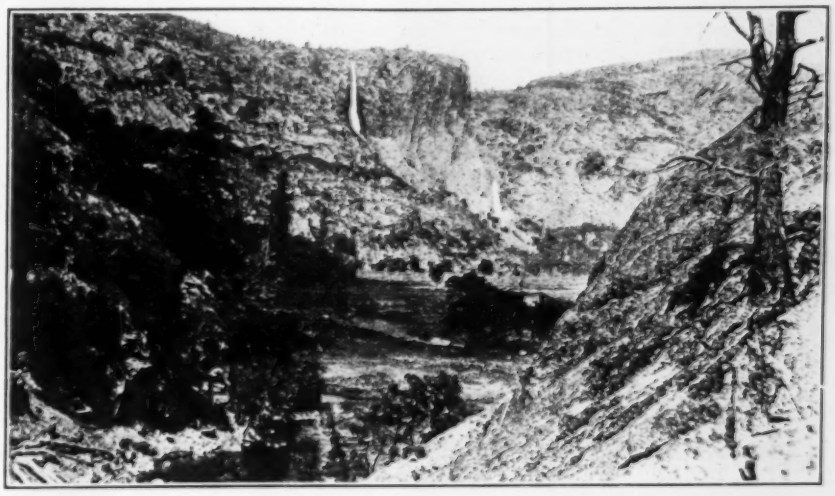


FIG. 13 BEAUTIFUL HETCH HETCHY VALLEY, THE SOURCE OF SAN FRANCISCO'S FUTURE WATER SUPPLY

Fig. 9 shows the interesting background of Big Creek Power House No. 8, in which may be seen the surge chamber and penstock of this great development.

Fig. 10 gives a view of the third 16,000-kw. hydraulic unit now being installed at Big Creek Power House No. 1. It shows the generator in the middle with the two overhung tangential water wheels plainly in view.

Fig. 11 shows the record-breaking banks of auto transformers at the Big Creek Power House No. 1, used to step up the voltage from 150,000 to 220,000 volts. Each bank is rated at 52,500 kva.

Fig. 12 indicates how the western engineer constructs a temporary lookout near installations of this character, in order that day by day photographs may be taken and a proper history of each day's constructive efforts may be recorded.

Often the mountain scenery encountered in this work is of an exquisite nature. Fig. 13 shows the famous Hetch Hetchy valley, where the city of San Francisco has been expending many millions of dollars in building a great dam near the tree shown in the foreground on the right of the picture. Today this great dam, together with the tunnels and the Moccasin Creek power house of 80,000 hp. capacity in the river 50 miles below, is nearing completion.

After a careful survey of developments now under way, the total program of hydroelectric development in the West as it has progressed from 1910 to date and may be expected to progress to 1930, may be summed up as follows:

	Horsepower Capacity		
Year	Hydroelectric	Steam	Total
1910	800,000	277,300	1,077,300
1911	886,910	324,690	1,211,600
1912	943,010	412,220	1,355,230
1913	1,051,540	467,251	1,518,791
1914	1,175,800	512,870	1,688,670
1915	1,332,430	542,580	1,875,010
1916	1,433,050	545,200	1,978,250
1917	1,566,390	554,680	2,121,070
1918	1,673,350	558,800	2,232,150
1919	1,701,546	619,344	2,320,890
1920	1,826,164	671,586	2,497,750
1921	2,116,500	742,350	2,858,850
1922	2,366,650	823,740	3,190,390
1923	2,728,670	861,330	3,590,000
1924	2,957,900	911,590	3,869,490
1925	3,210,040	943,760	4,153,800
1926	3,352,530	970,570	4,323,100
1927	3,630,280	973,260	4,603,540
1928	3,978,000	978,620	4,956,620
1929	4,177,740	978,620	5,156,360
1930	4,370,770	981,305	5,352,075

To indicate how undisturbed is the forward movement of hydroelectric development in the West, Fig. 14 is shown. By examination of this chart it will be seen that the upper portion of the black line, indicating the total power consumption in the State of California, has been at an even upward trend for many years past in spite of earthquake, pestilence, world war and financial depression. Hence we can predict with a considerable degree of accuracy the future of hydroelectric development in the years immediately ahead.

Two factors control this growth. Fig. 15 shows California's growth in mining, farm products and manufactures, according to figures from the United States census over a period of 40 years past. From this chart it is seen that farm products have long since outstripped mining. On the other hand, few have thought of California as an industrial state. It was as a consequence with much interest and surprise that its citizens heard recently of the

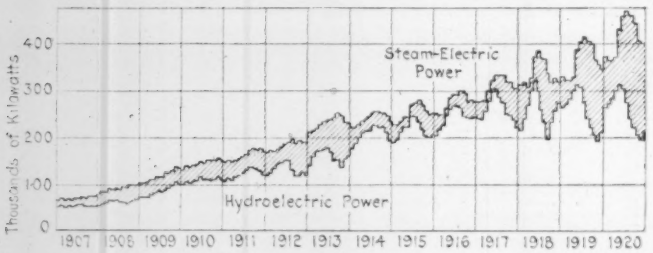


Fig. 14 Chart Showing the Steady Growth in Demand for Electric Power in California Since 1907

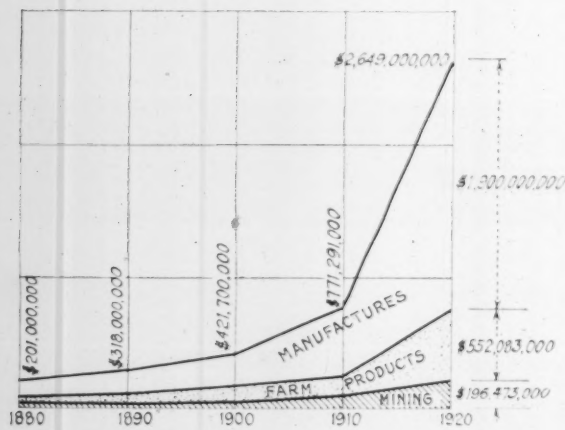


Fig. 15 Chart Showing Growth from 1880 to 1920 of Basic Industries Using Electricity in California

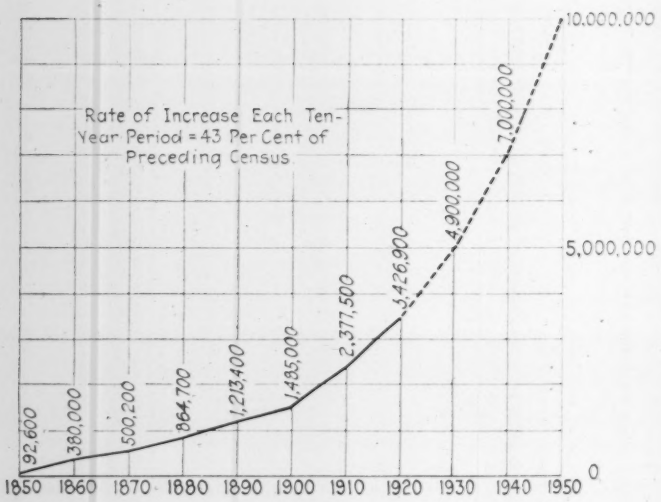


Fig. 16 Chart Showing Increase in Population of California Since 1850, and Predicted Increase for the Next 30 years

announcement by the Census Bureau at Washington that California had taken its place as eighth among the great manufacturing states of the Union, and was the fifth in rank as regarded the number of industrial establishments. In 1920 the value of California's industrial products amounted to one billion, nine hundred million dollars, and today their value is far in excess of this figure.

The other factor that controls California's growth is the matter of population. Fig. 16 shows her increase in population during the past 70 years together with predictions for the next 30 years. According to the census of 1920, California far outstripped any other state in the Union in this regard, her growth in population being 44.1 per cent. There is every reason to believe that, due to the present unexampled growth in the southern portion of the state, California is at present growing at an even more rapid rate; but assuming that her growth continues for the next 27 years as was maintained from 1910 to 1920, California will have a population in the generation immediately ahead of over 10,000,000 people; in other words, she will have a population by 1950 as great as we find today in New York, Pennsylvania, Ohio and Illinois.

A vision of the future takes us to the Pacific Ocean and the countries whose shores it washes, where now live over one-half of all the people of the world. With the vision of daring, enthusiasm, and constructive leadership in the developing of her great natural resources, the West has a destiny of service beyond comprehension, and with her vast waterpower resources she looks to the engineer for the upbuilding of a commercial and industrial empire such as the world has never before witnessed.

Small Electric Plants and Large Systems

The engineering reasons for the advantages of the grouping of many small plants into larger groups are generally known, but may be repeated. There are also similar large advantages to be gained from combining a number of small groups of plants to form a large *system* favorable to economic power supply, financing and management.

To use an illustrative analogy, quoting from the Economic Values of Electric Transmission by F. G. Baum, the large electric system may be compared to a bank in its economic function, and the electric medium of transmission may be compared to money, the medium of property exchange. If we had no money we would have to trade by direct exchange of property, and if we had money but no banks in which to deposit our funds from which those of us who need it might draw, the difficulties of doing business may be imagined. Without a central power-distributing system each consumer must develop his own power and have some surplus power. Hence, there is no medium of exchange, no means of "banking" the total power of all on the transmission system, and no drawing at such points and in such amounts as may be needed from time to time. Electric transmission provides the elastic medium for exchanging any one form of mechanical power to any other form of mechanical power at some point on the system.

Now if one hundred isolated power plants are connected together by a transmission system, and their power-producing possibilities thus concentrated on the system, we will then be able to supply the demands of the one hundred original consumers and have a fair-sized surplus always on hand. That is, by "banking" the individual energies through the medium of the transmission system, we can get all the convenience of the power facilities, and the system will be able to meet the demands of all its patrons—*and carry a surplus*. This is because we have concentrated the surplus of the several plants and because the individual needs do not all come at the same time. This surplus power is principally due to the fact that the diversity of interests now supplied from the one system is such that the system as a whole has a more uniform load, a better "load factor," than the individual plants when operating separately. In general, the more varied the interests which draw power from the lines the more uniform will be the demand, and the better for the transmission system.

Having now connected the hundred small plants by an electric-transmission system with a large power demand, we find we can no longer afford to operate the one hundred small plants, but instead, we install a few large power units to supply the entire system, and by this means produce a large saving in operating expense. The economies resulting from large power systems with large central stations in is in this way working a revolution in America, especially in California, and also in other countries. That this revolution is progressing from economic reasons which are fundamentally sound is certain, and therefore larger and larger electric power systems may be expected. (From Atlas of U. S. A. Electrical Power Industry by Frank G. Baum.)

Recent Developments of the Motor Coach

The Renewed Interest Taken in This Type of Equipment and the Reasons Why Railway Officials Have Been Forced to Turn to It—Advantages and Disadvantages of the Various Types of Gasoline-Engine, Storage-Battery, Steam and Gas-Electric Cars—Performance Data

By C. E. BROOKS,[1] TORONTO, ONT.

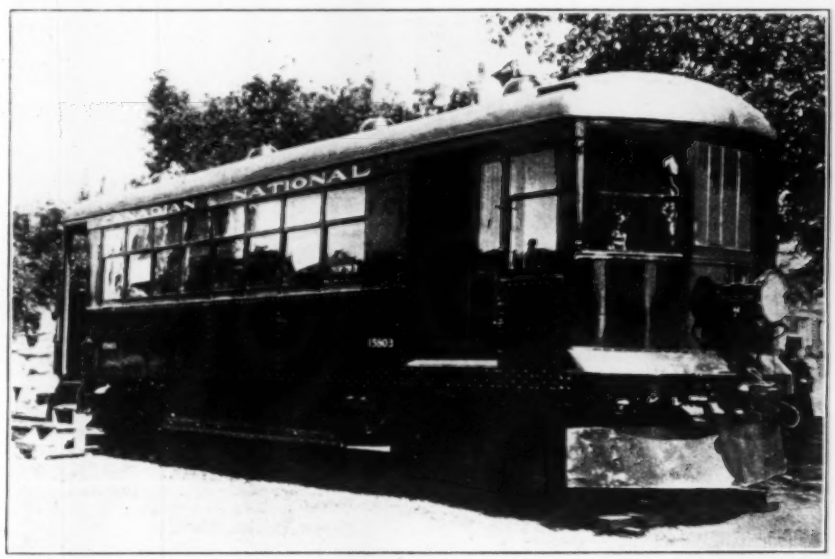
THE motor coach is not a new departure in railway work but is instead a branch which on most railways has sunk back from a promising start 15 years ago into obscurity due essentially to high maintenance and operating costs and to the general idea that the motor car was a toy only. During the past three years, however, interest in the subject has been actively revived as a result of

a The ever-increasing encroachment of the motor bus on railway earnings

b The improvement in the design of the motor and its appurtances, and

c The attitude of the public, which the automobile is rapidly convincing that plush seats are not necessary when traveling short distances.

In order to enter into a discussion of present-day developments it will be necessary to outline briefly the service requirements, and in doing so the author must be pardoned if he presents the motor coach as a "unit car" and not as a train of cars or a driving car hauling a trailer. The latter methods of operation may be possible on small interurban or suburban lines, but for practical transportation purposes they are not being considered by many large railways. The reason for this is that up to the present time it has not generally been found to be economical to replace a necessary train service with a motor coach hauling one or more trailers on account of motor-capacity limitations.



CAR FITTED WITH EDISON STORAGE BATTERIES

On the other hand, high train-mile costs for small returns have forced railway officials to turn to the motor coach for relief in the following classes of service:

a Giving a frequent passenger service on sparsely settled branch lines or parts of main line adjacent to market towns or junction points

b Connecting junction points on important main lines with the town or small city situated within a few miles of the main line

c Giving a group of towns situated on a main line or important branch lines a frequent connecting service over and above through main-line trains

d Connecting small summer resorts, golf clubs, etc., to branch-line or through main-line service

e Handling milk of a limited amount to a distributing or connecting point

f Providing connections to small suburbs.

Among the first real developments of the motor coach was the gas-electric car, the use of which spread rapidly fifteen years ago but which unfortunately did not solve these problems successfully on account of the high maintenance cost of the heavy-duty gasoline motor driving the generator, the very uncertain service, the complications of the equipment, and the great weight of the motor coach itself. Indications are that the gas-electric system with a modern gasoline-engine generating plant is making another bid for this class of work, and a brief description of such a unit is given further on in the paper.

About the same time steam units having very considerable weight and all the complications of the locomotive appeared, but were discarded for much the same reasons as the gas-electric car.

CLASSIFICATION OF PRESENT-DAY DEVELOPMENTS

Present-day developments are generally to provide for traffic requirements, which for the purpose of discussion are subdivided as follows:

Class A—A seating capacity of from 24 to 40 persons and provision for approximately 100 sq. ft. of baggage space. The light weight of such a car to be from 18,000 to 30,000 lb. maximum, or in other words, not to exceed 750 lb. per single seat (with baggage space) or 500 lb. per seat if no baggage space is allowed. In some classes of service the baggage space is given up and seating accommodation substituted.

Smaller cars have been made and are in daily use, but considering the rapid development of traffic after cars are put into service, it is believed that the above-mentioned limits are reasonable.

Class B—Units seating from 40 to 60 passengers and providing for baggage space a minimum of 100 sq. ft. and maximum of 200 sq. ft. As constructed these cars weigh from 800 to 1200 lb. per single seat, but it is thought that the weight must be kept down to the same limits as those prescribed for Class A in order to get fuel economy and keep maintenance costs at a reasonable figure.

Speed Requirements. Schedule speeds of 15 to 25 miles per hour, including stops. Consider a 5-mile run between stops and 1 min. for stops: Acceleration from 0 to 30 m.p.h. in 2 min. on level track, 30 to 35 m.p.h. at end of third minute, an average speed of 35 m.p.h. for the next $3^{3}/_{4}$ miles and 0.5 min. to travel 0.21 mile and come to a stop, means that in order to run 5 miles and make one stop the elapsed time is 10.9 min., or an average schedule speed including stops of $27^{1}/_{2}$ m.p.h. This allows nothing for loss of time on gradients, slow orders, etc.

Generally speaking, Class A units have been gasoline-driven, and the experience of many railways goes to show that this class of equipment is an economical and lasting development which will be improved to the point of high-grade automobile reliability within a very short time. Already in many places these cars have retrieved business which had been lost to bus lines on the highways and also to the privately owned car, and it has been the usual thing to find that passenger traffic develops to a marked extent after a service has been instituted.

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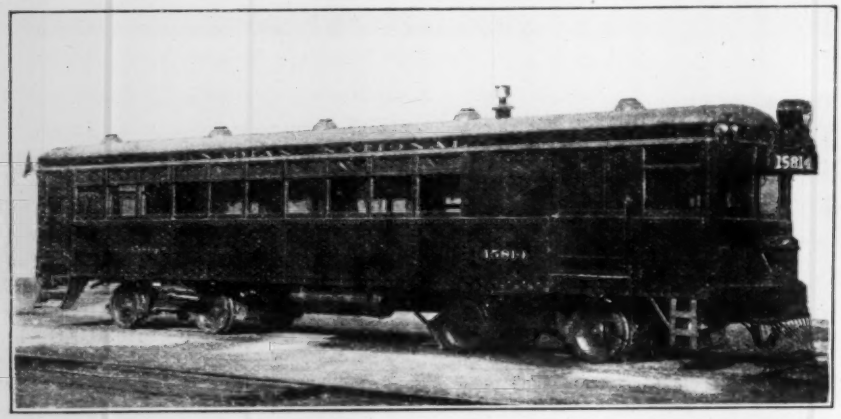
The failures have been rather heavy due to conditions described later in this paper, to the overexacting requirements of time tables, and to non-realization of the limitations of the gasoline motor.

For the purpose of description Class B units may generally be subdivided into (*a*) storage-battery, (*b*) steam, (*c*) heavy-duty gasoline-engine, and (*d*) gas-electric cars.

CLASS A CARS

The Gasoline-Engine-Driven Motor Coach. For Class A cars weighing approximately 30,000 lb. light weight, the general practice has been to use a high-grade 4-cylinder truck engine running at a maximum speed of approximately 1600 r.p.m. and developing a maximum of 70 hp. Wherever this type of engine has been used it has transmitted its power through clutches, transmissions, and universals to gears, most of which are of standard truck or even heavier design.



SERVICE CAR

A general description of a typical power plant such as mentioned above is as follows:

Engine, 4-cylinder, $4^3/_4$ in. by 6 in.
Pressure oiling system
Pump water-cooling system
Primary and secondary transmission
Primary ratio: first speed, 4 to 1; second speed, 1 to 1.

The secondary increases the ratio from 26 m.p.h. for normal engine speed to 35 m.p.h. The first provides for ruling-grade and the second for level-track conditions.

To a much less extent the automobile-type six-cylinder engine of the following general description has been experimented with:

Engine, 6-cylinder, $3^3/_{16}$ in. by 5 in.
Pressure oiling system
Gear ratio, 4.7 to 1 between engine and wheel
Nominal engine speed at 30 m.p.h., 1450 r.p.m.
Horsepower developed, 50 at 2200 r.p.m.
Maximum speed, 2200 r.p.m.
Pump water-cooling system.

In general its power has been transferred through standard automobile clutches, transmissions, etc., which are used with the same type of engine in automobile service.

So far as the actual power plant is concerned, it is the opinion of many that the automobile engine has in almost every way demonstrated its superiority over the truck engine for Class A cars and for general service because of its ability to run over rated speed without serious loss of balance and consequent excessive vibration, and its economy under light load conditions. The first of these reasons undoubtedly embraces conditions which are vital to the successful maintenance of any machine or engine, and it is the author's intention to attempt to explain this from the every-day point of view arrived at through experience rather than from the dynamics of the problem.

Practically any high-grade automobile engine designed for a rated engine speed of approximately 1450 r.p.m. at 30 m.p.h. with a gear ratio between 4 to 1 and 5 to 1 may be driven at engine speeds of 2200 r.p.m. and car speeds of from 50 to 60 m.p.h. without any noticeable vibration of a serious nature. Experience indicates that a similar flexibility cannot be expected from truck engines for any length of time without serious engine trouble developing and possibly resulting in a complete breakdown of the service.

When a motor coach is being operated on a railway where there are schedule connections to make and where there are meeting points designated by train orders and by the time card, it is certain that, regardless of the framing of a schedule, which should not develop an engine speed over that coinciding with the rated speed of 35 m.p.h., the operator will frequently exceed this by 15 or 20 m.p.h. in order to meet the requirements of the service after a delay of any kind. There is nothing parallel to this in highway work with either the automobile or the truck, but it is such an accepted fact on a railway that the only safe course is to provide the type of power plant which will meet these requirements daily without breaking down; in other words, the power plant must be moved out of the sphere of ordinary usage and into what might be called the outer edge of racing conditions. Railway gradients even accentuate this condition, as there is practically no opportunity for letting the engine cool off as there is on the highway.

The automobile engine has been designed not only for easy and economical low-engine-speed conditions, but also for those outlined in the preceding paragraph, and the experience of several railways in this country with a number of cars operating under extremely different conditions seems to bear this out, which leads to the conclusion that the light-weight, high-speed gasoline engine is a satisfactory power plant for the light-weight cars described as Class A.

CLASS B CARS

The Storage-Battery Car. The general data of a typical unit are as follows:

Car. Interior arrangement, to suit purchaser; weight, 60,000 lb.; length, 53 ft.

Trucks. Two 4-wheel standard M.C.B. axles, except that journals are fitted with roller or ball bearings.

Electric Motors, Etc. Four 25-hp. motors (250–300-volt) mounted with a gear ratio of 16 to 91. Standard series and parallel controller and circuit breaker installed at each end and in baggage compartment; provided with voltmeter, ampere-hour meter, underload circuit breaker and switches for control of air compressor and lighting.

Storage Batteries. 250 cells, capacity 450 amp-hr. at 300 volts, or 135 kw-hr. (580 amp-hr. have been obtained with a minimum of 150 volts).

Battery Charging. Direct current at 250 or 500 volts may be used for charging, and the car is equipped with switches for arranging the battery cells in either series or parallel. Normal rate of charging, 90 amp. Time required for a normal full charge, 5 to 7 hr. A higher rate of charging may be employed provided temperature of battery does not exceed 115 deg. fahr.

Radius of Operation. Maximum 140 miles on a full charge, figuring on level or rolling grade. Recommended not to exceed 100 miles without obtaining a boosting charge.

Power Consumption. Power required, 35 watt-hr. per ton-mile. Acceleration, $^1/_2$ m.p.h. per sec. Maximum speed, 45 m.p.h. on level track. As the car weighs approximately 60,000 lb., 35 watt-hr. per ton-mile is equivalent to 1.05 kw-hr. per car-mile for level track and normal conditions.

Within the above-mentioned radius of operation this car has been extremely satisfactory and is being operated successfully under low-temperature conditions with no appreciable trouble. Its tractive effort of 2400 lb. makes it possible to use a trailer if necessary. The cost of operation, including all maintenance and transportation charges, power, etc., is 17 cents per car-mile. The maintenance has been extremely light, and all indications are that the life of the batteries will be eight to ten years at least.

Steam Cars. The steam power plant was probably the first of any kind to be tried for self-propelled cars, but unfortunately its development has not kept pace with requirements. Medium-pressure (300 to 400 lb.) boiler plants with comparatively low superheat (100 deg. fahr.) were introduced to a considerable extent in continental practice several years ago, but the use of the steam car has not developed, due principally to the excessive weight and general complications of the equipment, and the inefficiency of the boiler plant.

Recent developments indicate that while the seriousness of these defects has been noted and improvements have been made, they have not yet been overcome to the point where steam power may

be considered to be the most satisfactory unit car power, and it may be well to consider these defects more in detail as they appear to exist in modern equipment.

Insufficient boiler capacity is a defect directly coupled with excessive weight of equipment. Boiler plants of approximately 70 boiler hp. nominal rating have been applied to cars with a total net weight in running order of from 50,000 to 60,000 lb. and providing for 100 sq. ft. of baggage space. The total live load will bring this equipment up to a gross weight of approximately 65,000 lb. or a load of 650 lb. per boiler hp.

Experience indicates that for a boiler of this capacity the weight should be reduced by approximately 15,000 lb., and the author has no hesitation in saying that this should be possible as the entire power plant may be carried on the leading (driving) truck and the remainder of the body lightened proportionately. Total absence of vibration should be a very great advantage in lightening the car equipment.

Practically all the first cars of continental make indicated that high pressure and high superheat were necessary in order to provide for a gear ratio sufficient for starting and at the same time for a piston speed of 800 ft. per min. at 40 m.p.h. The engine developed to meet these conditions has no doubt been a great mechanical success, but the boiler plant supplying it has not yet been developed to the point where it can exceed the schedule previously outlined, even under the most favorable conditions. The causes of this deficiency appear to be due to

a Insufficient header volume, resulting in carrying over an emulsion into the superheater units, with a consequent total loss of superheat and excessive water consumption, and

b Unequal distribution of heat to all generating units, resulting in steam pockets and thereby destroying both circulation and evaporative qualities.

The problem of providing sufficient surface-condenser capacity for hot-weather conditions coupled with protective appliances which will be operated by thermostat in cold weather has not been solved, with the result that under maximum conditions the water loss is as high as 45 lb. per car-mile, necessitating replenishing at frequent intervals and a consequent loss of time.

Automatic control of the oil flame has been highly developed but is not yet perfected. Generally speaking, this automatic feature must have two distinct functions: namely, (*a*) cutting off or reducing the fuel supply when maximum boiler pressure is reached, and (*b*) cutting off fuel under low-water conditions. The first-named is undoubtedly perfectly developed but the second is not, due to the varying quality of the steam coincident with the condition (maximum load) which most often causes low water.

The least considered of all conditions in connection with the steam car are probably those affecting the comfort of the operator, and in this respect it is unpopular on account of the extreme heat which may be experienced and also the noise of the oil flame. The latter is by far the most serious, and it is apparently impossible to control it when using high-velocity jets of steam for atomizing.

Space does not permit a further study of the steam plant, but the following general data of a steam car now being tested may be of interest.

Space required for boiler plant, sq. ft.	640
Heating surface of boiler, sq. ft.	385
Heating surface of superheater, sq. ft.	44
Gross weight of car, lb.	60,000
Net weight of complete power plant without oil and water, lb.	13,000
Quantity of water supply, U. S. gal.	200
Quantity of oil supply, U. S. gal.	180
Gallons of oil per car-mile, average	1
Boiler pressure, lb. per sq. in.	800
Superheat, average, deg. fahr.	200
Engine dimensions	6½ in. bore by 8 in. stroke
Gear ratio between crankshaft and axle	1 to 1.46

The Heavy-Duty Gas Car. The general data of a car of this type are as follows:

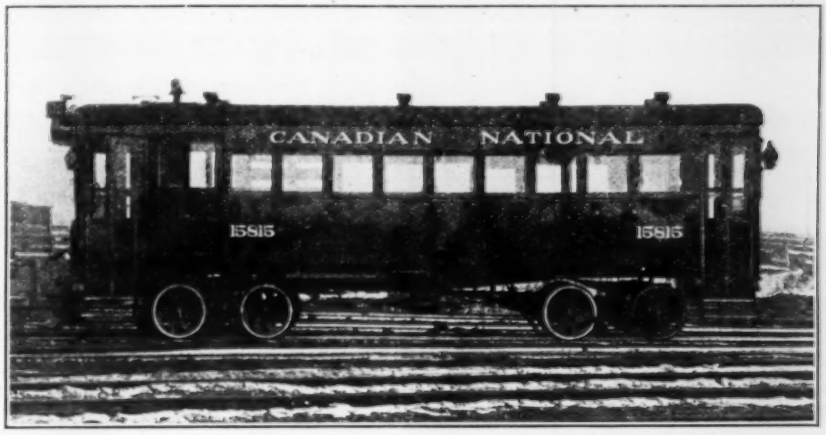
Car. Length, 55 ft.; weight, 66,000 lb. loaded.

Engine. 6-cylinder, 6¾ in. stroke. Power, 116 hp. at 800 r.p.m. and 225 hp. at 1600 r.p.m.

Transmission. Four speeds forward, three reverse; geared to give 56 m.p.h. forward at 1400 r.p.m. in high and 37 m.p.h. in third speed.

This class of car has not been tested to the point where any accurate data may be given, but it is evident that the gasoline consumption will be at least twice that of a Class A car per car-mile. The problem of handling through transmission and clutch the mechanical drive from a heavy-duty gasoline engine of possibly 200 hp. has not yet been solved unless it may be through the medium of the oil transmission so successfully used in navy work. However, the extreme complication of this transmission or magnetic control makes it doubtful at the present time whether gasoline power plants will successfully exceed 70 hp. in capacity.

The Gas-Electric Car. Mention has already been made of the car of this type which came into prominence about fifteen years ago and which was practically discarded on account of its unreliable power plant and general complication of equipment.



CAR EQUIPPED WITH SIX-CYLINDER AUTOMOBILE ENGINE

The first-mentioned cause of failure has undoubtedly been overcome and reliable constant-speed units are now in general use for generating purposes. If it were possible to eliminate the starting troubles when using types of engines suited for low-grade and cheap fuels, there would be no doubt about the general use of this type of equipment on account of unit power costs. The difficulties mentioned are such an important factor in the successful operation of a motor coach in certain localities that we must necessarily turn to the gasoline engine for generating power.

The gas-electric system provides double-ended control and an efficient starting torque but still retains all the complications of a dual power plant.

The general data of a modern gas-electric car may be stated as follows:

Car. Length, 55 ft.; width, 10 ft.; seating capacity, 54 with 100 sq. ft. of baggage space; weight loaded, 65,000 lb.

Engine. 6-cylinder, governor-controlled, 7 in. bore by 8 in. stroke; develops 150 b.hp. at 900 r.p.m. and drives a 100-kw. 700-volt generator which in turn drives two motors on forward truck. Fuel consumption (estimated), 0.25 gal. per mile.

It is thought that there may be a possibility of employing a smaller-capacity constant-speed gasoline engine (average running power consumption approximating 25 hp. for cars weighing 60,000 lb. loaded) which will drive a generator charging a limited battery capacity. Theoretically this might provide the starting torque desired and at the same time eliminate the undesirable features of the large power plant, but it could not be an economic consideration where cheap power could be purchased.

Ball and Roller Bearings have been one of the most important factors in the development of the motor coach. Exhaustive tests indicate that the ball bearing has reduced starting friction under summer conditions to approximately 15 per cent of that of plain bearings, or in other words has reduced friction of approximately 20 lb. per ton to 3 lb. per ton. At the same time the average rolling friction at speeds up to 30 miles per hour has been reduced by approximately 40 per cent, or from 3.6 lb. per ton to 2.2 lb. per ton.

Experience in this country indicates that the ball bearing is

suitable for Class A cars, but that the areas and sizes for designed industrial work should be at least doubled for railway work due to excessive shocks and side thrusts. It is not possible at the present time to say whether side thrusts are more destructive than vertical rail shocks, but it is certain that for poor rail conditions the bearings should have a side-thrust capacity of 100 per cent of the vertical load.

For Class B equipment it may be necessary to use roller bearings for vertical loads in connection with special bearings for side thrusts.

Transmission and Type of Drive for Gas-Driven Cars

While it has been stated earlier in this paper that the automobile engine is most suitable for the light motor coach, it must nevertheless be admitted that experience indicates that standard automobile transmissions, clutches, universal connections and driving gears are entirely inadequate for motor-coach service and are the cause of probably 75 per cent of the breakdowns. Similar parts which have been developed for truck engines are generally much superior due to their greater size and strength per horsepower transmitted.

The argument has been advanced that the parts are designed for the engine and will handle all the power developed by it; but this contention is not sound, because the greater inertia to be overcome at starting requires a momentary torque much in excess of anything experienced in automobile work.

Account must also be taken of the fact that in this country and the northern part of the United States cars have to be operated in snow storms, resulting in clutch slippage and shocks to transmission which are much in excess of those experienced in automobile service.

The method employed in transmitting the power to the wheels has generally been one of the following:

1 Through a standard transmission to one driving axle which supports the entire two-wheel rear truck and which can move in a vertical plane only. While this is the simplest method of driving, it is not the opinion that it will ever be generally acceptable to the railways as railway experience indicates that safety and good riding qualities are almost proportionate to the number of wheels in the trucks. This is particularly applicable to cars operating on cheaply maintained lines.

2 To both axles of the front four-wheel truck by gearing and universal connections from a transmission located behind the truck. Experience on some railways goes to show that this method is very successful, and although the number of universal connections is not reduced the shafts are all short and the driving forces are entirely removed from the passenger-carrying part of the cars, thereby reducing vibration.

3 To a transmission located at about the center of the car and from there to nearest axle of each four-wheel truck or to both axles of two-wheel trucks. The advantage claimed for this method is that the entire weight of the car is available to give good adhesion (where the trucks are of the two-wheel type), but general experience indicates that this is not necessary and is harder on the engine than an arrangement where part of the momentum of rotating parts may be taken up by slippage. Where four-wheel trucks are used this method has been found to give better adhesive qualities than connecting to both axles of one truck, but the advantage seemingly is not sufficient to warrant the extra complication of transmission.

4 To the leading axle of rear four-wheel truck. The chief advantage of the drive to the rear truck is that the engine may be aligned in such a manner as to have its shaft center line pass through the center line of the main driving axle, thus reducing wear on the universals and friction to a minimum. The disadvantage is that it necessitates the use of one or more supplementary bearings between the engine and the point where the drive shaft is coupled to the front universal. The maximum lateral motion of a truck of 48-in. wheel centers and 18-in. truck centers on an 80-ft. radius curve is shown by road check to average $^3/_4$ in. at a radius from the center of 24 in., so that it is apparent that the swing of the truck has but little effect on the universals. The torque arms supporting the housings of such an arrangement should have both vertical and lateral swing. Only when the load on the main axle is not sufficient for adhesion it may be conceived that driving power acts on the second axle. Under ordinary conditions transmitting power to the second axle generates no more friction than that due to the weight of rotating parts.

Methods of transmitting power from the front axle to the rear axle of rear truck may be subdivided as follows:

a Chain Drive. Chain drive has the disadvantage of rapid wear, noise, and the complication of shields and covers which more than overcome the advantage of the straight drive to the rear truck.

b Gear Drive. Gear drive to the second axle no doubt appears to be the best mechanical means of transmitting power, but it has all the disadvantages of rapid wear due to difficulty of adjustment of contact and the maintenance of extra universals.

c Side Rod and Cranked Wheels. Side-rod drive to the second axle, along with many other locomotive developments said to be crude and inefficient, in actual practice is a thoroughly reliable and easily adjusted and inspected arrangement, and is operating successfully at high and low speeds and with no appreciable friction.

d Miscellaneous, including oil transmissions that are still being experimented with.

Summary

The author realizes that no doubt a very great difference of opinion exists among railway men regarding what type of motor coach will fulfil the requirements of the service best. It is apparent to any engineer that there will be important developments in railway motor cars, both as regards design and field of service, and therefore the following conclusions may be considered as preliminary only.

Class A Cars (30,000 lb. max. light weight), driven by automobile-type, 6-cylinder gasoline engines, should have the following general characteristics and limiting conditions: 1.6 hp. per 1000 lb. light weight of car at engine speed of 2100 r.p.m., giving a car speed of 35 to 40 m.p.h. Rolling friction, 2.2 lb. per ton of weight on rail, and wind resistance of 5 lb. per sq. ft. cross-sectional area of car at 40 m.p.h. Weight of car per passenger seat, maximum 750 lb.; weight of car per maximum hp., 600 lb. Gear ratio between engine and wheel for ruling grades not exceeding 1.25 per cent, 4.7 to 1. (In hard operating conditions this may be increased to 5.5 to 1.)

Class B Cars. It is difficult to come to a general conclusion with regard to this class of equipment, as it appears that only one class of car has actually passed out of the experimental stage. One of the Canadian railways has found the battery car to be a thoroughly reliable and economical unit to operate, provided that the schedule will permit of charging time. Severe weather conditions have had but little effect on the operation, and the simplicity of the power-controlling devices eliminates any chance of opposition from the operators.

The chief obstacles in the development of this type of unit are the first cost and, in some regions, high power costs, but it is felt that the great advantage it possesses of double-ended operation without complication more than offsets any serious disadvantages.

In conclusion, it seems safe to say that there is a fairly large field for the motor coach in railway work and that it will not be developed from without but rather from within—slowly, conservatively, by motor manufacturers and railway engineers, as the traveling public will never tolerate from the railway companies the difficulties and disappointments which have been visited on them personally by the automobile manufacturers. The railway engineer to do his part in this problem must be familiar not only with operating conditions but also with the labor problems which are sure to arise in such a development, making it necessary and advisable to give consideration to the employee operating this equipment and to the public using it.

Discussion

L. Klopman,[1] who opened the discussion, said that he agreed with all that the author had said in favor of the storage-battery

[1] Gen. Mgr., Railway Storage Battery Co., New York, N. Y.

car, but that the paper did not go quite far enough in presenting the proved utility and service of the storage-battery unit.

The author had divided self-propelled cars into two classes placing the storage-battery car in the heavier class (B) but excluding it from the lighter (A), or that for cars having a seating capacity of 24 to 40 passengers, a baggage space of approximately 100 sq. ft., and a car weight of from 18,000 to 30,000 lb. maximum. Storage-battery cars answering this description had been in successful operation since 1917 on the Chattahoochee Valley Railway in Georgia, on the Long Island Railroad in New York since 1914, and elsewhere, which had a seating capacity of 32 to 40 passengers, a baggage space of 80 sq. ft., and a weight, including battery, of 28,000 lb. These cars had been equipped with two 25-hp. 175 to 250-volt motors, double and series-parallel control with a gear ratio of 5.29 to 1, and 150 A-12 Edison cells.

There was no reason why a storage-battery car such as this could not meet every test which had been successfully met by the gasoline-engine-driven motor coaches designated by Mr. Brooks under Class A, and not only meet these conditions, but more economically and with greater reliability in operation than the gasoline car.

It was a matter of common knowledge that the ordinary overhead-trolley car stayed in constant daily service with little or no repair so far as the motors were concerned for periods as long as 20 years.

Assuming the average daily operation of a car in the Class A group to be 100 to 125 miles per day, it was well known that in interurban work the trolley motor performed this service day in and day out, year after year, with only nominal motor repairs and practically no "out of service" conditions.

It was equally well known that no automobile engine would do 100 miles per day continuously, 36,500 miles per year, without considerable repairs and replacement of parts. All automobile owners knew the necessity of frequent regrinding of valves to insure good operation, to say nothing of replacement of piston rings, pitted cylinders, worn gears, transmission shaft, etc., which meant a car frequently out of service for many hours. With such engine equipment the liability existed of complete power failure, and when this happened it necessitated at great expense, annoyance and delay to traffic, substitution by steam service.

The multiplicity of troubles and complete failures which were inevitable with any kind of combustion engine and drive after considerable service necessitating shopping of equipment and consequent substitution, were matters worthy of serious thought. It was generally conceded that the second-hand value of an engine which had run 36,000 miles, or about the equivalent of one year's continuous daily operation, was a very nominal amount, which indicated the large depreciation per annum which must be charged against such a type of car.

This was to be compared with the simple, effective electric motor and control of the storage-battery car, which was as nearly absolutely foolproof as any mechanism yet devised. The four-motor drive permitted in the remote contingency of failure of even two motors, of the remaining two motors' propelling the car. A mechanical or electrical failure of the battery in this service was so remote as to require no consideration. Its initial cost was admittedly higher than that of a gasoline tank, but in a comparatively short period of operation the low cost of maintenance both for the battery and the electric motors soon equalized the difference in initial cost.

The life of the storage battery of which he was speaking had proved to be over ten years in this class of service and, in fact, cars were now in operation in Canada the batteries of which had been installed as far back as 1914. The condition of a gasoline engine started in the same service in that year could be easily imagined.

It was to be remembered that all storage-battery cars were equipped for double-end control, which meant both economy of time and money in operation. The author had cited the use of these cars to connect junction points, small summer resorts, golf clubs, etc. This service would hardly warrant the cost of installation of a Y or turntable such as the one-end operation of a gasoline motor car made necessary, and it would easily be seen that it was only fair that the cost of such installation be deducted from the initial cost of storage-battery operation for both Class A and Class B service.

So far as Class B cars were concerned, he would call attention to the fact that in stating the maximum radius of operation, the author had limited himself to the statement that its maximum radius of operation was 140 miles on a full charge. This statement was correct as far as it went, but in Canada, as well as elsewhere, cars of this size had made and were making up to 235 miles a day where facilities were available and operating schedules permitted the time for intermediate charging. This would give the storage-battery car a mileage which, for all practical purposes, was equal to that afforded by a gasoline-engine-driven car.

R. G. Gage[2] said that one of the chief items that should be watched in connection with storage batteries was the first cost—not perhaps in the car itself, but in the different facilities for making low cost of operation. The charging facilities sometimes were different to procure. The question of operation of the storage-battery car in Canada was important. At certain times of the year, with the temperatures which they had, it was absolutely necessary to start the car out properly charged, but in such cases it had been his company's experience that there was very little trouble. That was one of the serious things that must be given consideration as regarded the proper scheduling, which those having to do with transportation were not always very ready to realize.

J. A. Shaw[3] stated that the railway company with which he was connected had installed four gasoline-engine-propelled cars, which were still in service, to transport passengers from the station to its hotel. They operated for four months each summer and carried all the supplies and fuel as well as passengers. By their installation the use of 200 horses had been done away with and the cost of gasoline for operation per annum was about $1000, which was small.

He had investigated a number of cases where electric cars could be operated. The street-railway company approached was willing to furnish the power required at a very low rate, with practically no charge for auxiliary equipment.

The author, in closing, said that the chief reason why he believed that the gasoline-driven car was the most suitable to fulfil the requirements outlined for Class A cars was the requirements of the service.

These small units were often used in very sparsely settled parts of the country where the only hope railway companies had of making anything out of the service was in its frequency. His company had driven cars fitted with automobile engines on regular schedules which necessitated covering 360 miles per day, up to 50,000 miles before any appreciable repairs had been necessary to the engine.

It was necessary, he said, to differentiate between experience derived from highway-run service and mechanical equipment driven on steel rails. He felt sure, from what he had seen, that the life of a gasoline car motor running on steel rails would be five or more times that of a gasoline automobile motor on highway service.

It was necessary to remember, the author said, that in Canada, where car batteries had to be imported, their cost would run close to $20,000. If necessary, the complete gasoline power plant in one of the Class A cars could be junked every year for practically the interest on that amount alone, without repayment of the cost of the batteries. It might appear at first that it was wise to figure on writing off the value of an automobile engine driving one of these small units after, say 50,000 miles. Experience had indicated, however, that it would be proper to allow 150,000 to 200,000 miles on that kind of service. But even if conditions were such as to make it advisable to junk the engine at the end of 50,000 miles, he still thought it would be better to do this rather than to incur the very heavy expense for the storage battery, which necessitated a large investment in the charging equipment where cheap power could not readily be tapped. He appreciated the value of the storage-battery car, but felt that for certain classes of service it was necessary to have something a little cheaper and a little more flexible.

[2] Signal and Elec. Engr., Canadian National Railways.

[3] Elec. Engr. Canadian Pacific Railways.

Modern Hydraulic Turbines of Large Capacity

A Discussion of the Subject with Special Reference to Refinements in Design, Increased Efficiency, Improved Test Methods, and Advances in the General Art Which Make the Use of Large Turbines Possible

By H. G. ACRES,[1] TORONTO, ONT.

ON NEW YEAR'S DAY of 1905, in the plant of the Canadian Niagara Power Company, it fell to the author's lot to bring up to speed, for the official inspection of the Provincial Park Commissioners at Niagara Falls, Ontario, two 10,000-hp. turbines, the largest units ever built up to that time. Almost exactly seventeen years later it was again his privilege to turn over the first of the five 55,000-hp. turbines now installed in the Queenston plant of the Hydro-Electric Power Commission at Niagara, the largest-capacity units at present in operation. It was this more or less fortuitous combination of circumstances which suggested a subject for discussion.

Not in any sense as a precise definition, but for the purpose of placing a reasonable limit on the scope of discussion, a hydraulic turbine of large capacity may be defined as one having a capacity of not less than 25,000 hp. and a revolving weight, with the generator rotor, of not less than 200 tons, exclusive of hydraulic thrust.

The above limitation on capacity and revolving weight will serve the double purpose of eliminating from the discussion any consideration of low-head installations and at the same time giving prominence to certain problems which become increasingly serious as the operating head ranges upward from 70 or 80 ft. As units of the class which has just been defined above are rarely installed under heads as low as this specified minimum, the discussion is therefore more or less limited to medium- and high-head plants.

Where a high operating head is involved, there is always present the problem of controlling high static pressure in the interests of safety; and in most cases there is also the problem of controlling and utilizing the flow characteristics of long pipe lines to insure not only safe but practicable operation. Finally, where a copious water supply is associated with a high head, there enter the problems of safely and efficiently controlling a heavily energized water column, and handling the immense revolving weight of the modern super-turbine.

Static-Pressure Control

Control of static pressure involved in super-turbine water supply is a vital factor in relation to a maximum degree of safety and convenience in operation, and in continuity of plant output. Obviously the mechanism to exercise this function is a valve, and such a valve, for super-turbine service, must have the following general specifications:

a It must be built in any size necessary to accommodate any conduit, however large.

b It must be designed to operate under any static head, however high.

c Its operation must at all times be safe, sure, and positive, in opening under full static pressure, and in closing under the most serious plant emergencies, such as the breakage of a wheel case or a runaway resulting from failure of governor control.

d It must open and close rapidly, and should have an automatic closing feature to meet the above-mentioned emergency conditions.

Obviously such a specification could not be met by any valve of the ordinary type, nor was it met until the advent of the Johnson automatic plunger valve. This is now an indispensible adjunct to super-turbine installation, and constitutes one of the advances in the general art which contributed to the possibility of present-day super-turbine development.

Flow Control

In the majority of cases the development of a high head involves the use of some type of closed pressure conduit of considerable length. In the case of an automatically governed plant where the ratio of head to length of conduit is more than about 1 : 10, certain phenomena become manifest which constitute a serious menace to safe and practicable operation. The continuous rejection and pulling on of load increments causes recurrent surges throughout the length of the conduit, which, if not damped out or relieved, will sometimes multiply and become superimposed to such an extent as to cause pressures enormously in excess of the static head. This condition, in conjunction with the conversion of pressure into velocity head for acceleration purposes, when a large increment of load is pulled on will impose a duty on the governors which they are not intended to perform, and where long closed conduits are used with turbines gated beyond the point of maximum efficiency, as they nearly always are, a condition may obtain which is entirely beyond the regulating function of the governor.

The power-discharge curve for the ordinary turbine will show a maximum production per second-foot at the point of best efficiency and the first derivative of this curve will show a rapidly decreasing rate of production per second-foot from that point to full gate. Consequently, every horsepower pulled on in excess of best-efficiency capacity will require a rapidly increasing amount of water to produce it. If, therefore, the turbine is operating at or beyond the point of maximum efficiency, when the system demands a large increment of load there will be a falling off in production per unit of water supplied, and coincident therewith a loss of effective head, due to the absorption of such velocity head as is necessary to accelerate the water column. It is therefore evident that during the period when the water column is accelerating, a condition may obtain where the power output is actually falling off while the water input is increasing, the result being that the governor may open the gates full stroke in response to the falling speed, at which point its controlling function will cease, until either the system wr^2 holds the unit, with dropping frequency, for the space necessary for the governor to gradually resume control on rising speed, or until the generator drops its load and the turbine jumps to runaway speed at full gate, with possible disastrous consequences.

In so far as the control of surge pressures is concerned, it is evident that if no practical limitations are placed upon it in the matter of diameter and height, the ordinary standpipe would be an ideal corrective agency. It is clear, however, that in the case of a super-turbine installation unrestricted scope in design is wholly impracticable from the standpoint of both cost and space limitation. On the other hand, a simple standpipe, feasibly designed and located, may at times actually make more acute the conditions it is designed to correct or alleviate. This is due to the fact that the simple standpipe can only passively absorb surge pressure, with its recurrent phases, and for this reason often acts as an agency for superimposing, one upon the other, pressure waves generated by successive changes in load.

Another obvious expedient under this head is to limit the range of velocity change in the conduit by means of a synchronous bypass, actuated by the governor mechanism. This contrivance can be adjusted to prevent the occurrence of disruptive pressures in the conduit, and also to supplement the influence of the system wr^2 in holding the unit within the regulating range of the governor when the unit is pulling on or rejecting load. In this connection it is to be noted that the condition under discussion has not necessarily to do with the maximum conduit velocity at any one time, but with the *range of change* in conduit velocity over a short period of time, and with the absorption and building up of head energy induced by these changes, within their low and high limits.

[1] Chief Hydraulic Engineer, Hydro-Electric Power Commission of Ontario.

Presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers. Slightly abridged. All papers are subject to revision.

The synchronous bypass therefore has a useful function, but in the case of a high-head super-turbine installation its usefulness would be largely discounted by its cumbersome dimensions, its waste of water, and an added mechanical complexity which should be avoided whenever possible as a matter of principle.

The discussion under this head has now reached a stage where it is possible to define the specifications of an ideal surge-control agency.

a It should have the effectiveness of a simple standpipe of very large dimensions, without its cost and space requirements.

b It should be capable on the one hand of preventing or counteracting the effect of any undue absorption of head energy when load is pulled on, and, on the other hand, of preventing or counteracting the effect of any serious recurrent surge pressures arising from load rejections, whether isolated or successive.

c The water surface should be sufficiently active to prevent freezing in ordinary low temperatures, and the dimensions should be such as to allow feasible and effective frost protection against extraordinarily low temperatures.

d It should operate without wasting water.

e It should be mechanically simple, with a minimum of moving parts and adjustments.

f It should be entirely dissociated from the governor mechanism, leaving the governor free to perform its own peculiar and highly important function, which is to control the speed of the generator, and not the vagaries of the water column.

This specification has been met with a large degree of effectiveness by one of the most wholly original and at the same time most useful contrivances ever developed in the field of hydraulic engineering, namely, the Johnson differential surge tank. Ample published data are available concerning its structural details and operating principle, and it is only necessary to state here that this contrivance constitutes another of the essential advances in the hydraulic art which have made possible the development and safe operation of the super-turbine.

REVOLVING WEIGHTS

The handling of the revolving weight of a modern super-turbine is governed largely by consideration of efficiency, a matter which will be discussed later, only the mechanical aspect of the problem being considered at this time.

In the early days of hydraulic-turbine development the step bearing, a form of combined thrust and guide bearing located under the runner, was used almost exclusively for the support of revolving element. The subsequent gradual increase in the efficiency, speed, and capacity of turbine runners then began to introduce problems of pressure intensity, depreciation, and accessibility, which the designers of that day solved, not by improving the step-bearing principle, but by abandoning it and inaugurating, as a result, the era of the horizontal-shaft turbine. This led to advanced development in bearings of the pillow-block type, together with the introduction of marine-type thrust bearings to take up unbalanced runner thrust, a double requirement which gave rise to serious problems when the development of electrical generation and transmission called for continuously increasing turbine capacity and speed.

The original turbines installed in 1896 by the Niagara Falls Power Company represent the first reversion to the primitive basic principle. They were the super-turbines of that period, and were of the vertical-shaft type. The revolving weight of these units was partially suspended from and partially superimposed on a step bearing located above the runner and immediately below the generator, thus removing one of the main disadvantages of this type of bearing, that of inaccessibility. A further important innovation was the application of external pressure to a film of oil which was forced in between the moving and stationary elements of the bearing.

The oil-pressure thrust bearing had a vogue of many years' duration, and some bearings of this type are still in operation. Their disabilities are: expensive investment and maintenance cost, mechanical complexity, and high temperature, resulting in low oil viscosity and high energy losses. Also, even a momentary failure of the pressure-oil supply usually results in the loss of the bearing.

Finally there came the ultimate conception, based on the simple and true embodiment of a basic principle first established by the experiments of Tower in 1883 and afterward mathematically demonstrated by Reynolds. It remained for Kingsbury to exemplify this principle in a mechanism which is one of the most outstanding examples extant of the simple and efficient application of a natural law.

The Kingsbury-type thrust bearing has the following characteristics which distinguish it from its oil-pressure prototype and make it an eminently suitable mechanism for supporting, in motion, the revolving weight of the modern super-turbine:

It can be adapted to safely support the revolving weight of the heaviest turbine, within feasible dimensional limits.

The source of oil supply is static, and integral with the bearing itself.

The oil supply is "unlimited" in the sense necessary to conform with the laws of motion of viscous fluids, as enunciated by Reynolds.

The formation of the "pressure wedge" is not induced by any external agency but by the motion of the bearing itself, and by providing for a very slight lack of parallelism between the stationary and moving elements.

The wedge pressure is a direct function of the speed of the moving element, and the thickness of the oil film is a direct function of the degree of viscosity of the fluid, and consequently of fluid temperature, which can be regulated within any desired limits by the simple expedient of water-cooling coils.

Apart altogether from their very material contribution to operating efficiency, the mechanisms above discussed introduce a factor of safety and dependability, either singly or in combination, without which the present-day status of super-turbine development would have been unattainable.

REFINEMENTS IN DESIGN

The items above classified as essential advances have been radical departures from contemporary practice, and have actually opened up the possibilities of super-turbine development, whereas the turbine governor has been enlarged and improved more or less synchronously with the enlargement and improvement of the hydraulic turbine, and to meet the constantly increasing importance of the duty it has been called upon to perform.

Governors. As the primary requirements of governor operations are precision and reliability, the natural trend of design has been in the direction of simplicity in principle, the reduction of lost motion by the use of minimum of moving parts, and precise shop work. An interesting development along this line is the White shaft governor, where the centrifugal element is attached direct to the main turbine shaft, thus obviating the complication of a belt or gear drive for the flyballs.

The double compensation principle, with the restoring mechanism and load-limiting attachment, appears to meet all requirements of super-turbine governor control adequately and safely, and speed regulation is assured within safe operating limits for all gate movements up to the full stroke of the servomotors.

Remote switchboard control has also been developed to such a stage that there is considerable latitude as to the location of the actuator, which in some cases in now placed directly over the servomotor cylinders, in others on the machine-room floor, and sometimes on the switchboard gallery.

A useful adaptation of the Johnson valve principle has been devised by Taylor for interchanging governor and hand control. Two sets of plunger valves are used, one set on the governor and one on the hand control, and the throwing over of one hand lever will simultaneously close all the governor valves and open all the hand-control valves, or vice versa. This operation can be carried out in such a short interval of time that there is little or no chance of losing control of the turbine, and no chance whatsoever of interfering with the proper sequence of operation.

One of the latest innovations in governor design is a motor-driven centrifugal element, the motor being located on the flyball spindle and synchronously actuated by the main generator. This device serves the same purpose as the shaft governor in that it eliminates the belt or gear drive.

Draft Tubes. The activity which has manifested itself during the last four or five years in draft-tube investigation is directly

traceable to the vogue of the super-turbine. There are two main reasons for this: first, the large concentration of power in a super-turbine permits an appreciable increment of useful power to be reclaimed as the result of a gain of only a fraction of one per cent in overall efficiency; and second, the vibration resulting from vortex action and unsteady vacuum becomes magnified in the super-turbine to such an extent as often to cause serious inconvenience in operation. The pioneer development under this head was the White hydraucone, which marked a distinct advance in this branch of the art.

Briefly, the twofold object of recent draft-tube experimental research has been to regain as much as possible of the whirl component of the velocity head in the runner discharge, and to devise means of so training the combined axial and whirling flow, through appropriately designed water passages, as to minimize conditions

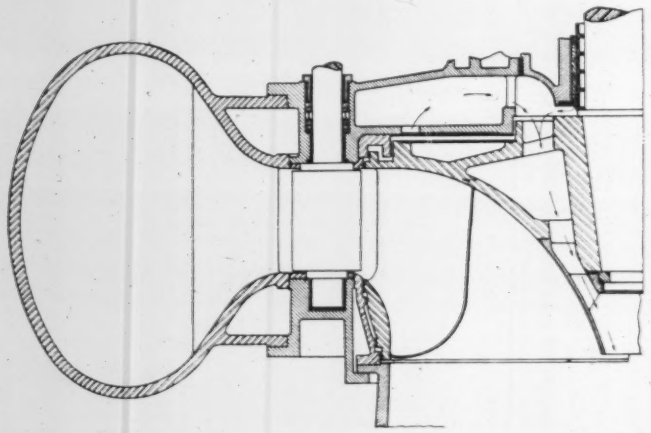


Fig. 1 Showing Application of Labyrinth Seal and Overn Disk to Hydraulic Turbine for Reduction of Leakage

giving rise to vibratory effects. All experiments carried on along this line by the principal turbine builders have definitely proved the inadequacy of all forms of the elbow-type tube, and have developed a series of types, more or less correlated, which have given satisfactory results when tested under operating conditions. Of these types the Moody spreading tube seems to conform most nearly to one's conception of the conditions to be met. This is particularly the case in respect of a cone in the center of the Moody tube which extends from the invert to the lower extremity of the runner hub. Inasmuch as this cone solidly fills the region of maximum turbulence, it seems entirely reasonable to assume, even without experimental confirmation, that it will be effective in preventing cavitation and the formation of vortices, with their resultant vibratory effects. Serving this purpose it also naturally follows that it must affect a useful conversion of the hitherto wasted whirl energy of the central zone, realizing thereby a gain in efficiency as well as a betterment in operating conditions.

Leakage Prevention. In a small high-head plant with which the author had something to do the turbine showed 85 per cent efficiency on acceptance test. A year later the efficiency had dropped to 67 per cent by reason of excessive leakage through the runner clearance spaces. The head in this case was 550 ft. and the circumference of the clearance spaces was about 14 ft. In the case of the 55,000-hp. turbines at Queenston the head is about 250 ft. less, but the circumference of the runner clearances is about 65 ft. It is evident, therefore, that in the case of a runner of this size, under such a head, the leakage factor is a serious matter. As a matter of fact, two of the most recent refinements in turbine design have been devised for the express purpose of meeting this condition: namely, the so-called "labyrinth seal" for preventing leakage through the runner clearance space, and the Overn disk for preventing leakage through the gate clearances.

Instead of the ordinary simple seal consisting of a straight annular passage past the crown of the runner into the space under the head cover, and a similar passage past the runner band into the draft tube, the labyrinth seal, Fig. 1, consists of a series of alternately expanded and contracted passages which destroy the velocity head and reduce the head on the final free jet to one-third of its initial value.

The Overn disk, Fig. 2, functions by introducing into the clearance space between the end of the gate vanes and the distributor plates an obstruction to the leakage flow considerably greater than the diameter of the gate shanks, thus effectively reducing the leakage at these points.

Taylor Sectional Scroll Case. Where the head is sufficiently high to require the use of a cast-steel scroll case, a serious problem is introduced which has to do with the cantilever strain on the radial joints between the speed ring and the scroll case. With the ordinary design, efficient bolt distribution is not practicable, and in the shop pressure test, where the scroll is not supported by a surrounding mass of concrete, the bolts are sometimes stressed beyond the elastic limit, with resultant serious leakage. This disability can be largely overcome by casting the speed-ring stay vanes integral with the scroll sections, but the pouring and annealing of such a casting is a most difficult piece of work and the results are not always certain. The Taylor sectional scroll case is just as effective and largely removes any uncertainty as to the quality of the casting. Taylor's method, see Fig. 3, is to cast the speed-ring stay vanes integral with a small radial section only of the scroll, and connect later by means of a joint to the main section of the scroll. This joint is so located as to permit of a heavy flange and efficient bolt spacing, and the castings are of such shape and dimensions that there is every assurance as to their quality after leaving the annealing furnace.

The showing made by the sectional scroll case under shop pressure test indicated that no undue risk would be involved in setting the scroll in the open, with only so much concrete as might be necessary to secure a stable anchorage.

With regard to the pitting and erosion of runners, it may be said that super-turbine development has now reached a stage where this condition is no longer primarily a problem of design but of economics. In other words, if the customer so specifies, the manufacturer can select a specific speed and supply, at a price, a turbine in which the runner will have as long a period of useful life as the other major elements of the installation.

Such a specification on the part of the customer would, of course,

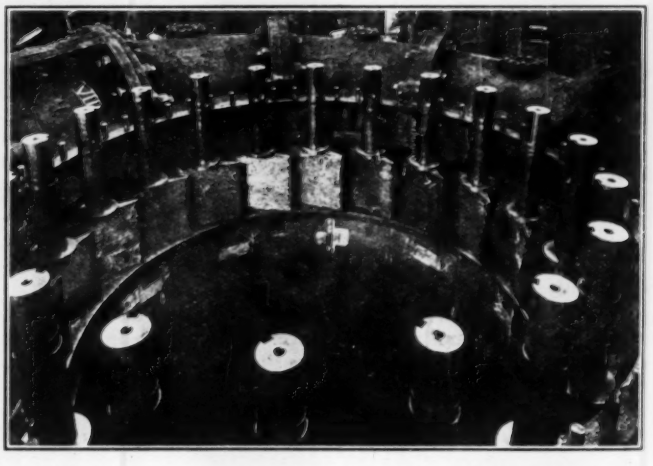


Fig. 2 Showing Application of Overn Disk to Guide Vanes for Reduction of Leakage

involve additional capital expenditure for the generator as well as for the turbine, but as against this aspect of the situation it must be realized that the modern super-turbine frequently has a capacity which enables it to earn upward of $2500 every twenty-four hours. Consequently the lost revenue charge against runner replacement in a fully loaded unit may easily run as high as $25,000. If this were necessary every two years, it would be equivalent, on a 6 per cent basis, to a capital charge of $200,000. Such being the case, it is evident that the choice of a proper economic specific speed is a matter deserving the most careful and mature consideration, being a factor of at least equal importance with proper gateage and elevation relative to tailwater.

EFFICIENCY

The overall maximum efficiency of the Queenston units is well beyond 90 per cent, and within their maximum efficiency range they deliver about 32 e.hp. to the switchboard for every second-foot of water supplied under 305 ft. of net head. Under such conditions a variation of one per cent, one way or the other, on one of these 55,000-hp. units would mean either the lack, or the availability, of sufficient power to meet the requirements of an average community of 2000 population. This statement should serve to emphasize the significance of high efficiency as related to super-turbine practice.

The primary factor making for high efficiency is a minimum of obstruction to the direct flow of water from forebay to tailrace. This consideration involves water passages of ample section, with changes in direction of flow eliminated wherever possible, and where unavoidable, careful proportioning and transitioning. These requirements constitute the general definition of high efficiency, and are exemplified in the carefully designed annular water passages in the Johnson valve, in the wheel case, speed ring, runner, and draft tube of the modern super-turbine, and also in the total elimination of one change in direction of flow entering the wheel case, by vertically suspending the revolving weight from a Kingsbury-type bearing. Secondary but none the less significant factors are the elimination of leakage, low power loss in thrust and guide bearings, and expert shop work.

The proper handling and correlation of these various factors is exemplified in the results illustrated in Fig. 4, which shows the efficiency and power-discharge curve for one of the Queenston turbines. The matters of interest in connection with these curves are as follows:

1 The maximum efficiency is 93½ per cent.

2 The efficiency at the point of maximum discharge is 88 per cent.

3 The turbine has a capacity range of 32,000 to 63,000 hp. at efficiencies of 90 per cent or over, and a capacity range of 37,000 to 60,000 hp. at efficiencies of 92 per cent or over.

Fig. 5 illustrates the overall switchboard efficiency curve of the same unit, and indicates a maximum overall efficiency of 91 per cent.

The outstanding fact in connection with these results is that the modern super-turbine is capable of converting into mechanical

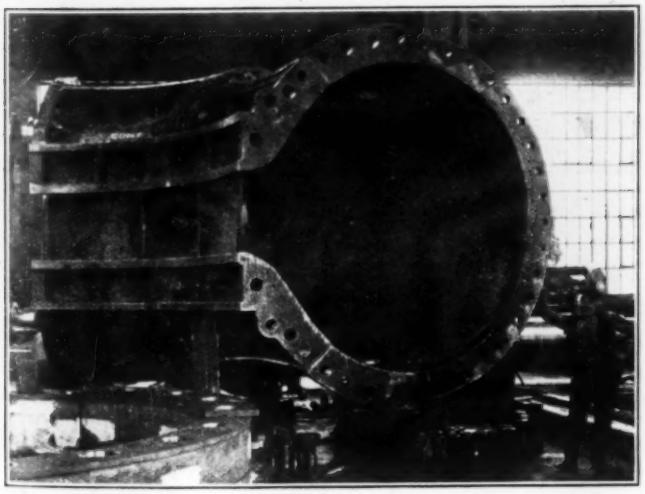


FIG. 3 SECTION OF TAYLOR SECTIONAL SCROLL CASE SHOWING INTEGRAL CAST STAY VANES AND JOINTS SUBDIVIDING SECTIONS

energy all but 7 per cent of the gross potentiality of the water supplied. Extreme conservatism in the fixing of specific speed, and the combined effect of further small refinements in the design of water passages, may possibly raise this efficiency another per cent in the future, and may give the curve a slightly more advantageous shape, but it would really appear that the super-turbine of the present day embodies the ultimate in respect of water economy at the point of best efficiency.

Fig. 6 is an interesting curve representing the first derivative of the power-discharge curve of one of the Queenston turbines. As its name implies, this curve is derived by plotting the amounts of horse power per second-foot produced, in various regions of gate opening at and beyond the point of maximum efficiency, by a very small opening movement of the gates. In other words, it shows, not the gain, but the rate of gain in power for increasing water input.

It will be seen that when operating at the point of maximum

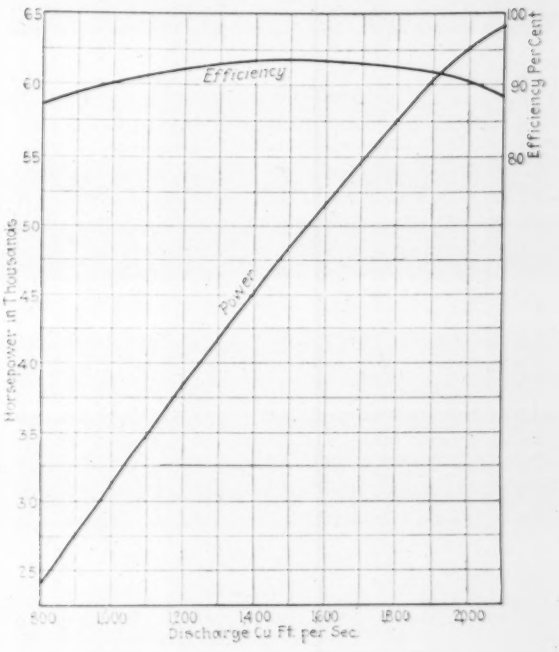


FIG. 4 POWER-DISCHARGE AND EFFICIENCY CURVES OF ONE OF THE QUEENSTON TURBINES

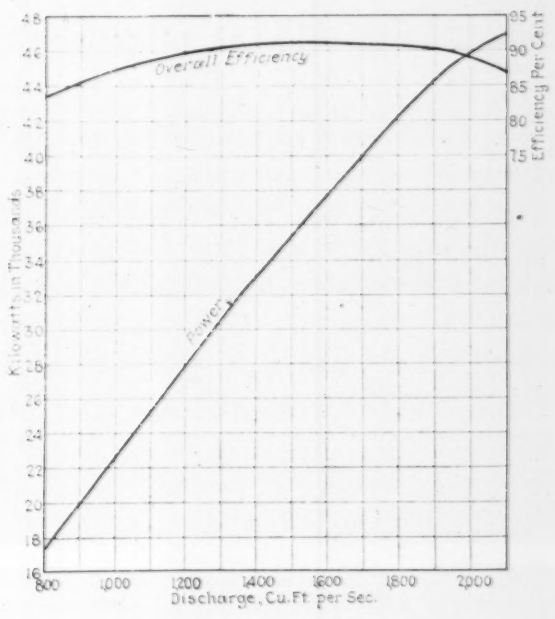


FIG. 5 OVERALL EFFICIENCY AND POWER-DISCHARGE CURVES OF UNIT OF FIG. 4, HEAD, 305 FT.

efficiency with a water input of 1600 sec-ft. the gain in power for one additional second-foot supplied is 32 hp. On the other hand, at full gate, when the turbine is taking water to the extreme limit of gate opening, the gain in power for one additional second-foot supplied is only about 10.5 hp.

The two extremes above cited serve to illustrate the significance of previous statements made under the head of flow control, and also to confirm the truth of the following statement: that under high heads, where the gross potentiality per second-foot of water

is correspondingly great, it is in the best interests of economy, as well as safety, to operate normally at the point of best efficiency, and to employ the excess capacity of an overgated turbine for emergency purposes only, and not for routine operation. This statement has an added significance, also, when the water supply is artificially stored.

Furthermore it is obvious that as the point of maximum efficiency is also the point of minimum hydraulic loss, the rate of runner deterioration at this point must also be a minimum. From this point on the rate of runner deterioration is an accelerating progression, directly related to the degree of overgate.

The points in favor of gating the super-turbine back to the point of best efficiency are: first, safe and efficient operation, particularly where long pressure conduits are involved; second, economy in the use of water under high heads, where the gross potentiality per second-foot of water is great, and where artificial storage is a factor; and third, on account of runner deterioration, particularly where runner replacement involves a material revenue loss.

The matter of turbine gateage has not, in the past, received the

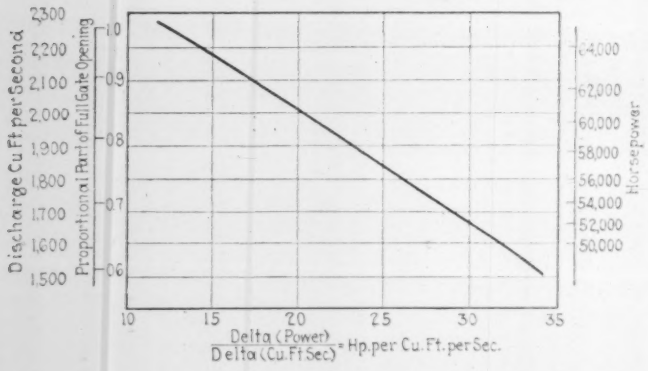


FIG. 6 FIRST DERIVATIVE OF POWER-DISCHARGE CURVE OF ONE OF THE QUEENSTON TURBINES, SHOWING RATE OF GAIN IN POWER FOR INCREASE IN DISCHARGE

attention it deserves at the hands of the purchaser. It does not particularly interest the manufacturer, and must be covered, if at all, by the customer's specifications.

It would also appear that the main factor offsetting the above argument is one of cost and not of engineering: namely, the additional cost, if any, of providing emergency plant capacity in the form of a separately installed unit or units, instead of relying on the overgate capacity of the normal installation.

It may be stated in conclusion that generalizations have probably less weight in hydraulic work than in any other branch of engineering, as nearly every prospective installation is a unique problem in itself, and no class of construction work is less governed by precedent.

TEST METHODS

The factors entering into a turbine test are forebay and tailwater level measurements, net head measurements, records of gate opening, and measurements of power and flow.

Most of these factors are susceptible of easy and accurate determination by well-established methods, but in the case of flow, the immense quantity of water taken by the modern super-turbine introduces an almost insurmountable problem as regards accurate measurement by ordinary means. It so happens, however, that there has recently been devised a method of flow measurement in closed pressure conduits, which by reason of its accuracy, cheap application, and conformity to an established principle of natural law, entitles it to be classed among the essential advances in the hydraulic art as related to super-turbine practice.

The Gibson Method of Flow Measurement. This method of flow measurement, known as the Gibson process, is based in principle upon Newton's second law of motion, and upon the less generally known theorem of Joukovsky, which is to the effect that when the velocity of flow in a closed pressure conduit is retarded, an oscillatory pressure wave is induced, the intensity and amplitude of which is proportional to the degree of retardation and to the duration of the period over which the retarding influence acts. Briefly, Gibson used the penstock and turbine gates to produce the Joukovsky pressure wave, which he recorded graphically with an apparatus of his own devising, and then reduced the result by the mathematical process of Newton's second law.

The method is therefore almost unique in the field of practical hydraulics in that it involves no empirical constants whatever, and the mathematical accuracy of the result is limited only by a relatively small instrumental and personal error involved in graphically recording and measuring the pressure wave.

The Gibson process has been fully described in the technical press, and it is not the intention here to cover the details of its theory or application. It may not be amiss, however, to describe briefly, the unique and interesting method of obtaining and using the "pressure-time" diagram which forms the basis of computation.

The recording apparatus consists of a mercury U-tube connected to the penstock through a $^1/_4$-in. pipe. The glass leg of this U-tube is connected to a camera box containing a lens focusing on a light-proof cylinder, the latter carrying a sensitized film and revolving at constant speed behind an oscillating seconds pendulum. When gate closure occurs, the mercury column in front of the lens rises and falls with the pressure wave, and the record is printed on the revolving film. At the same time the stem of the seconds pendulum, swinging across the face of the lens, records the time period of gate closure on the film, in the form of a vertical black line.

Fig. 7 illustrates a typical pressure-time diagram thus obtained. The point *A* represents balanced conditions in the system just previous to gate closure. The distance from *A* to the vertical line *EF* represents the full-time period of gate closure. The oscillations to the left of the line *EF* are due to the mercury column gradually stabilizing by friction, until at the point *P* static head is registered, with the penstock water column at rest. The vertical distance between the point *A* and the point *P* therefore represents, to the scale of the diagram, the recovery of velocity and friction

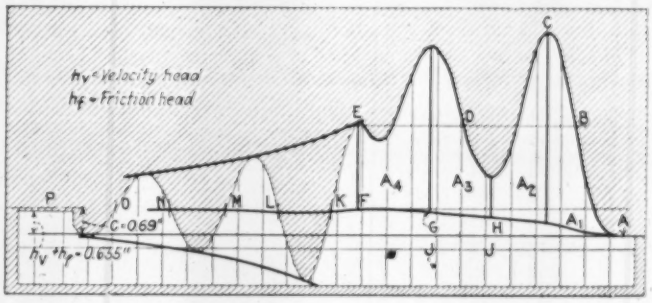


FIG. 7 TYPICAL PRESSURE-TIME DIAGRAM OBTAINED IN FLOW MEASUREMENT BY THE GIBSON PROCESS

head. The lines *J* represent the second intervals recorded by the pendulum.

The total area above the base line, and to the point of gate closure at *F*, represents a total energy absorption containing three separate elements, velocity head, friction head, and the destroyed momentum of the water column. This latter element being the one required for the application of Newton's second law, it follows that the energy absorption due to recovery of velocity and friction head must be segregated.

At the point *A* it is evident that the full value of velocity and friction head is existent, while it is equally obvious that at *F*, the point of final gate closure, this energy has been fully recovered. The line *AHGF*, technically known as the "recovery line" divides the total energy area into two parts, the upper part of which, *ABCDEFGH*, represents the value sought, namely, the destroyed momentum of the water column. The area below the recovery line of course represents the sum of the other two elements, friction and velocity head. Intermediate points between *A* and *P* on this line are obtainable through the fact that the area generated at the end of any partial period of gate closure is proportional to the amount of flow reduction at that point, and also that the velocity and friction heads still unabsorbed are proportional to the square of the residual flow. The intermediate points *C* and *H*

were determined on the basis of these relationships, through the medium of the measured sub-areas A_1, A_2, A_3, and A_4.

A test of one of the Queenston units, with all the other necessary operations in connection with head and gate-opening measurement, etc., was made complete within about four and one-half hours, and included 28 separate runs at various gate openings, an average of ten minutes per run. The unit itself was out of commercial operation for about one hour altogether, showing clearly the facility and cheapness with which the Gibson test can be carried out.

An interesting feature of these tests manifested itself in an apparent inconsistency in some of the finally computed points for the efficiency curve, which varied for equal power outputs. This trouble was due to a discrepancy in the power readings, which were actually correct in themselves within the error limits of laboratory standards. The reason is that under the Gibson process the velocity measured is that existing at the instant the gates begin to move, and if the speed of the unit is perfectly uniform at this instant, the corresponding power reading will be correct for the measured discharge. If, however, at the instant of the flow measurement, the unit is in process of a speed change even so small as to be unnoticed on the frequency meter or tachometer, an error will be introduced in the electrical power measurement which the Gibson process is sufficiently precise to detect.

The flywheel effect of the Queenston generators is 21,000,000 lb-ft.[2] If, at the instant of a flow determination, the speed of the unit is dropping at even so small a rate as one cycle in ten seconds, there is 570 hp. registered on the wattmeters which the wr^2 of the generator is supplying, but not the water as measured.

Conversely, if the speed of the unit is in process of increase at the rate of one cycle in ten seconds, there is 570 hp. which is not being registered on the wattmeters, but which the water, as measured, is actually supplying, for absorption by the generator wr^2.

This means that for any one value of discharge there may be any number of wattmeter measurements which may possibly be in error in any amount up to 1000 hp. It is possible that this disability may be overcome by devising some simple means of recording the minute speed change of the unit synchronously with the instant of flow measurement, and thus permit a proper correction to be applied to the power readings.

The Salt-Velocity Method of Measuring Flow. The use of the Gibson process is naturally limited to conditions which are properly conducive to the production of the phenomena upon which the theory is based, the first essential being a closed pressure conduit of reasonable length and uniform section. In short conduits of, say, 50 to 60 ft. in length, more especially if they have at the same time a non-uniform section, it becomes necessary to apply correction factors which have a greater relative influence on the ultimate accuracy of the result than is the case with long conduits of uniform section. It so happens, however, that the recently devised Allen "salt-velocity" method of measuring flow is susceptible of application to short as well as long conduits and to non-uniform as well as uniform sections, the accuracy of the final result being dependent, not upon an instantaneous pressure rise, but on the refinement of method and money outlay applied to the injection of a salt solution, the time of passage of which, through a fixed length of conduit, forms the basis of the final computation. Practical applications appear to indicate that it has a useful function and will become a recognized process as related to this branch of hydraulic engineering.

In conclusion, it may be well to call attention to the fact that the definition of a turbine of large capacity, as given in the preamble of this paper, is applicable in particular to the Francis type, as there are now in operation, under heads of less than 70 ft., super-turbines which would come within the definition as regards capacity and revolving weight. These turbines are, however, of the new propeller type and any discussion of this new departure in turbine design would open up a subject far beyond the scope of this paper.

In evidence before the International Joint Commission two years ago the author advocated the propeller-type turbine for the St. Lawrence River Development, and was subjected thereby to severe criticism. However, recent performance, under actual installation conditions, has largely confirmed his views and it is more than likely that the Development, when consummated, as it ultimately must be, will see the propeller-type turbine installed.

Discussion

Written discussions of the paper were submitted by Messrs. R. W. Angus, H. Birchard Taylor, and Lewis F. Moody, extracts from which follow. Professor Angus[1] wrote that much of the paper would apply with equal force to medium- as well as large-sized units, and much of it to low-head as well as to high-head plants. Surges were present in all plants having long closed supply conduits, and their magnitude depended on the friction in the line and on its length and the initial and final velocities in the conduit, as well as on the diameters of the pipe and tank, but was independent of the head on the plant, so that the control of surges was really of more vital importance in low-head than in high-head plants.

The form of draft tube, Professor Angus wrote, was not as important in high-head plants as was generally supposed. In the modern low-head turbine of high specific speed the water entered the top of the draft tube with a high velocity of whirl, and the ordinary bent tube was unable to regain efficiently the energy in the water leaving the turbine. The water in such a tube whirled as it passed around the bend and the discharge conditions were bad. In this case the tube must have a special form, capable of recovering the energy leaving the wheel, and the hydraucone and spreading draft tube appeared well suited to this purpose.

On the higher-head plants wheels of comparatively low specific speed were always used, and it was a relatively easy matter to deliver the water to the top of the draft tube in an axial direction at the load showing highest efficiency, and if this was done the straight tapering tube, or even the quarter-turn, gave very good results. It was true that there would be whirling of the water at all other loads but one, and the spreading tube might show higher efficiencies on these loads than the quarter-turn tube, but as yet there did not seem to be enough evidence available in this connection.

Leakage was a serious matter in high-head plants and, according to the illustration given by the author, was relatively much more objectionable in the high-head, smaller plants than in the larger ones. Water-turbine builders were trying the labyrinth packing which had long been in use on steam turbines and turbine pumps. It was true that in both the latter cases the rings had been used on smaller diameters than on water turbines, and this fact made the use of the labyrinth more problematical in the case of water turbines, although it seemed to offer a good solution of the leakage problem.

In regard to the making of reliable tests, there was serious danger in hurrying the work too much, and it did not seem desirable to shorten the tests unduly. Averages taken over a reasonably long time with steady loads, Professor Angus believed, would obviate the difficulty mentioned by the author, as the latter stated that errors up to nearly 2 per cent were possible.

H. Birchard Taylor[2] wrote that the Queenston project, which had first come to his notice in 1911, had been carried out substantially in accordance with the original program with respect to the unit capacity of the turbines, and that practically every improvement that had been developed in the art of turbine design and construction for high-head turbines had been incorporated in the units there installed.

When units of such large capacities as 60,000 hp. were adopted, the importance of such questions as the attainment and maintenance of high efficiency, the elimination of runner corrosion, the avoidance of dangerous water hammer or vibration, the prevention of permanent injury from wear and tear by the provision of renewable parts, and careful study of operating requirements, became evident. The author had shown the great monetary loss involved in only a few days' shutdown of a unit, and his figures made it clear that an enforced interruption of service would not have to last long to produce a loss exceeding the entire cost of a turbine unit. Any safeguard against such interruption, therefore, would easily justify a large capital outlay.

Still larger units were now being constructed for the Niagara Falls Power Co., namely, of 70,000 hp., and it would appear that

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[1] Prof. M. E., Univ. of Toronto, Toronto, Ont., Canada. Mem. A.S.M.E.
[2] Vice-Pres., Wm. Cramp & Sons' Ship & Eng. Bldg. Co., Philadelphia, Pa. Mem. A.S.M.E.

the factors which would put a limit on power capacity in the future would be other than those involved in the construction of the turbine itself; and it was reasonable to believe that units of larger capacity than any yet built, when feeding large systems, would be just as dependable as units of smaller capacity had been in the past when connected to systems of lesser magnitude.

Lewis F. Moody[3] wrote that he believed that any one who read the paper or investigated the manner in which the engineering of the Queenston plant had been carried out would be impressed by the unremitting endeavors of the author and his staff to secure the best possible solution for every element in the plant design, and by their adoption of new solutions for many problems arising whenever a new departure could be demonstrated to have advantages, even if its adoption was entirely contrary to accepted standards. Certain features, such as Mr. Taylor's method of constructing the turbine casing without a separate speed ring, had been worked out for the Queenston plant for the first time. Even in the features which had been previously tried out, however, an additional problem had been introduced at Queenston due merely to the unprecedented size of the turbines.

In his specification for the penstock valve for installations of this magnitude the author enumerated the most vital points which had to be covered. A supplementary feature which should also be considered was that of tightness. If the penstock valve passed any appreciable amount of leakage, it was unfitted for location close to the turbine for the purpose of permitting quick access to the turbine without unwatering the penstock. Another important feature was that the motion of the valve should not require the dragging of an unlubricated surface tangentially over another under a heavy normal pressure caused by the static head. The Johnson valve met both of these additional considerations, as the sealing took place by a motion at a large angle to the sealing surfaces, and only at the time of final closing did a heavy pressure due to the static head force the surfaces against each other, thus effecting extremely tight closure.

In connection with the author's references to the design of spreading draft tube adopted for the last three units at Queenston in which a central core was extended all the way to the runner, Mr. Moody called attention to comparative tests that had recently been carried out for the U. S. War Department at the I. P. Morris laboratory in Philadelphia. The turbine tested was of very much higher specific speed than the units at Queenston and consequently involved a more severe draft-tube problem due to the relatively much greater tangential or whirl component present in the water leaving the runner. A draft tube which successfully met the requirements of this high-specific-speed turbine, however, would be equally applicable to the low-specific-speed turbine, and its ability to regain energy of whirl would be effective in improving over-gate and part-gate performance in a lower-specific-speed installation. The particular form of spreading draft tube showing the highest efficiency in these tests was one in which the central core had been carried to the runner as in the Queenston units. The tests had been undertaken primarily to learn whether it would be possible by a simplification of design or the use of an elbow form of tube to reduce the cost of the power-house substructure sufficiently to balance the loss in efficiency. They had shown that the increase in turbine efficiency secured by using the spreading type of tube was so substantial that its use was fully justified. It might be of interest to mention that this type of tube had been adopted for installation either built or building which aggregated approximately 1,750,000 hp.

With reference to the author's statement that the two 10,000-hp. turbines installed by the Canadian Niagara Power Co. in 1904, were the largest units ever built up to that time, he would say that the first 10,500-hp. turbine at Shawinigan was completed at practically the same time, and that it deserved a special place in this branch of engineering because of its being entirely of American design and because of its many years of successful service.

H. A. S. Howarth,[4] who opened the oral discussion, dealt at some length with the development of the Kingsbury type of thrust bearing and its characteristic features, as well as the influence which the work of Prof. Osborne Reynolds, had exerted on thrust-bearing design. The Michell bearing, which had frequently been described in the English technical press, was of the Kingsbury type and was independently invented by an Australian engineer who acknowledged indebtedness to Professor Reynolds' mathematical treatise.

William White[5] said that he was heartily in accord with what the author had to say in reference to specific speeds. Electrical engineers insisted upon going higher in speeds, which meant a greater liability of pitting. The author was the first representative of the customer that he had heard say that the customer also assumed the liability for some of the changes or damages which necessarily would result in going to these higher speeds.

He appreciated the author's remarks concerning the hydraucone and he wished to pay a tribute to Mr. Moody's work. However, he would call attention to the fact that the hydraucone and the spreading draft tube were one and the same thing, because tests had shown that when the cone was taken out the results were the same. Mr. Moody's work, however, had brought out the fact that the cone steadied the performance of the unit, and its extension up to the hub of the runner might prove a valuable addition.

He appreciated Mr. Moody's reference to the Shawinigan unit, as he had been responsible for its design.

Prof. C. M. Allen,[6] who devised the salt-velocity method of measuring water flow, gave brief particulars regarding it. In this method, which is said to be a very accurate one, a charge of brine is injected under air pressure into the stream flow and its passage between two or more known points is accurately timed. By dividing the volume of the conduit between these points by the time of passage, the rate of flow is obtained. The conductivity of the water varies as its salt content, and the timing of the passage of the charge is effected by means of a stop watch or recording seconds clock and indicating electrical instruments, or by recording electrical instruments.

In his closure the author, replying to Professor Angus, said that it was quite true that surges in low-head installations and draft-tube conditions in low-head installations had not been given much prominence in the paper. The reason was that under low-head conditions these two factors became, in the majority of cases, the major elements of design, and it had been impossible to include an adequate discussion of low-head plants under this particular subhead.

Professor Angus had mentioned the possible interest which might attach to a discussion of the comparative efficiencies obtained on the units at Queenston, using the spreading draft tube as against the elbow type. There was a considerable amount of interesting data available in connection with these comparative tests, but it had not yet reached the stage of coördination and final consideration where it could be profitably discussed.

With regard to the question of error introduced into discharge measurements, it was quite true that in the paper he had given figures that would lead to the inference that error was possible. As a matter of fact, this phenomenon did not occur every time one made a test run for discharge. He had merely mentioned that fact as an interesting feature brought out by a method of discharge measurement that at the present time was sufficiently accurate, and the existence of this error could be more or less counteracted either by multiplying the number of runs for any particular opening or by devising some simple method of keeping track of the minute speed changes of the generator at the time of flow measurement.

Mr. White had spoken of the comparative efficiencies of the hydraucone and the spreading draft tube with the cone, and he thought what he said was quite true in so far as it affected one factor, namely, the overall efficiency. He had been discussing that overall efficiency in so far as it was influenced by draft-tube design, and it was not possibly quite as important as the manifestations which gave bother in the form of vibratory effects. At the time Mr. White made his investigations those vibratory effects had not reached the stage of seriousness which they had at the present time. A decade ago they were simply taken as a matter of course, because they caused no trouble. Under present conditions, due to resonance, etc., they constituted a serious factor in operation.

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[4] Gen. Mgr. and Ch. Engr., Kingsbury Mch. Wks., Philadelphia, Pa. Mem. A.S.M.E.

[5] Mgr. Ch. Engr., Hyd. Dept., Allis-Chalmers Mfg. Co., Milwaukee, Wis. Mem. A.S.M.E.

[6] Prof. Hyd. Engrg., Worcester Polytechnic Institute, Worcester, Mass. Mem. A.S.M.E.

A Practical Laboratory and Drawing-Room Course in Industrial Engineering at Cornell University

By MYRON A. LEE,[1] ITHACA, N. Y.

The paper shows how the early courses in industrial management have been widened in scope to include the engineering phases of the problems of manufacturing. This has been accomplished by supplementing the lecture courses with a practical laboratory and drawing-room course in which the student has his own problem to work out in the design and operation of a modern industrial plant.

A NUMBER of years ago it became evident that the work of the engineer was broadening in scope. The engineer in practice was increasingly concerned with management problems, and it was deemed advisable to give the engineering student some instruction along the line of industrial management. The first attempt at Cornell was by means of a lecture course which proved very popular and was of unquestioned benefit to the student.

The increasing importance of this study to the engineering student was soon evident and the limitations of a lecture course were realized. It is recognized, for instance, in teaching machine design that a lecture or even a recitation course alone cannot accomplish results that can be attained in the drawing room. If a student actually designs some piece of apparatus such as a triplex pump, for example, he acquires and, what is more important, retains a knowledge of machine design which he could obtain in probably no other way. Moreover the knowledge he obtains is not limited in its application to merely the apparatus he has designed, but to a large extent is fundamental and applies to the whole field. This is particularly true if the drawing-room course is correlated with a carefully prepared and well-presented course of lectures which show the application of the principles of the subject to the general field.

It was felt that these well-recognized principles of teaching would apply equally well to industrial engineering, and that a course in which the student applied the principles of industrial management to definite problems would give him a knowledge of the subject which he would retain and could apply to any problem in that field. After several years of experiment and development, a course has been evolved which it is the object of this paper to describe.

Time Study

One part of the work consists in actual time-study practice. This work has no direct connection with the rest of the course and could be given at any time. As a matter of convenience it is usually undertaken during the early part of the second term. Studies are made by the student of actual operations in the university shops. As it has not been found practicable to have the student make a preliminary study, he is given mimeographed sheets which enumerate the elements of each operation to be studied and the time when each reading is to be taken. Before taking the study he is required to demonstrate that he has practically visualized the sequence of the elements making up the operation and will almost automatically know when to take readings. This requirment is the result of sad experience as it has been found that otherwise the student will be confused and will get no dependable readings for the first few runs. It has also been found that it pays to have the student practice reading the watch before he attempts observations on a workman. The standard time-study watch reading hundredths of a minute is used and a person accustomed to seeing an ordinary watch needs some practice before he can quickly read one of these watches, as he must when using the "continuous" method of observation.

Each student is provided with a stop watch mounted on a board of convenient shape and records his observations on the usual type of time-study log sheet. It is possible usually to have not more than ten men taking observations upon one operator. Several complete operations or cycles are observed so that representative times can be obtained.

After the observations are made the student then determines "reasonable minimum" times for each element, and making the customary allowances for personal needs, usual delays, etc., prepares an instruction card for the operation determining standard time for a given number of parts, and also writes a descriptive report of the study.

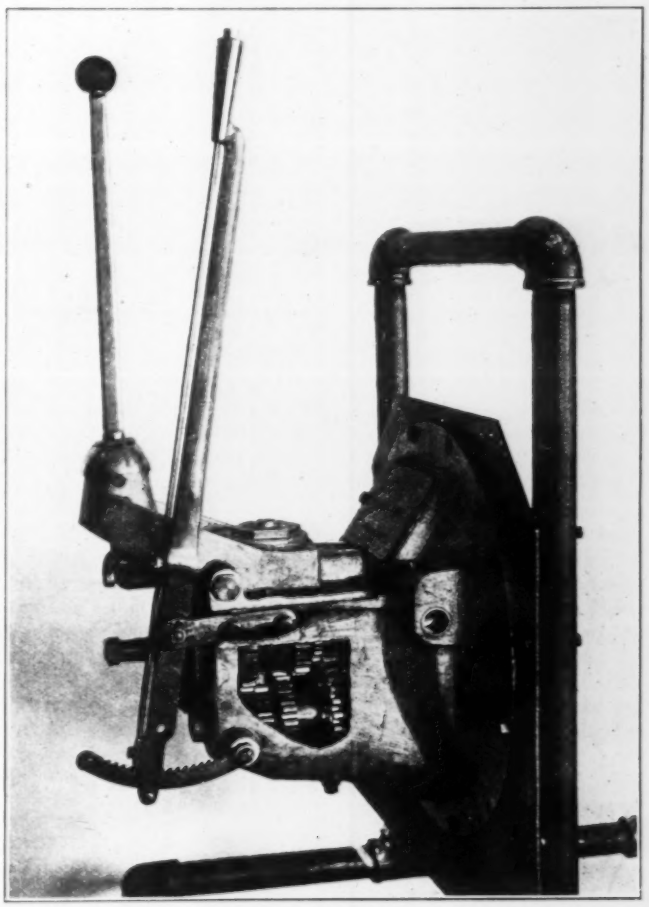

Fig. 1 Automobile Transmission Used as Basis of Student's Mass-Production Problem

Each student makes a time study of operations on perhaps five or six different jobs such as operations on a molding machine, a drop hammer, drills, turret lathes, etc. This work, of course, does not and is not expected to make an experienced time-study man of a student, but it does familiarize him with the taking of a study and stimulates him to a much keener interest in the lectures on the subject which explain the various methods, difficulties, and results of time studies.

A Practical Drawing-Room Course

In the endeavor to develop a satisfactory drawing-room course it soon became evident that specific and definite problems must be used. This was particularly true since the educational problem was one of mass production, a large number of students taking the

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course. As the problems to be taken up were to apply to factories it seemed advisable for each student to have (on paper) a factory to which his problem could apply. Through the courtesy of the Brown-Lipe Company, of Syracuse, blueprints for one of their transmissions and operation sheets for each part were obtained, and each student makes floor plans of a factory to manufacture a definite number of these transmissions. All students can make use of the same data, but since each is designing a plant

PART NAME Countershaft Drive Gear PART NO. 1
MATERIAL 3½% Nickel Steel
NUMBER OF PARTS PER TRANSMISSION 1

OPER. NO.	OPERATION NAME	MACH. NO.	MACHINE NAME	NUMBER PIECES PER CYCLE	NUMBER MACHINES RUN BY ONE MAN	CYCLE TIME, MIN.	CHANGE TIME, MIN.	SET-UP TIME, HR.
	Inspect Forging							
1	Bore & Face Both Sides	1	J & L	2	1	5.00	-	4.0
	Inspect Bore & Face							
2	Grind Sides	2	Grind	1	1	0.50	0.1	2.0
3	Turn O.D.	5	Fay	20	2	8.00	1.5	2.0
	Inspect O.D.							
4	File Corners	48	S. Lathe	1	1	0.25	-	-
5	Block Teeth	18	B & C Hob	16	2	13.00	5.0	3.0
6	Finish Cut Teeth	19	Shaper	5	2	11.00	2.5	3.0
	Inspect Teeth							
7	Broach 6 Grooves	3	Broach	4	1	1.00	0.25	3.0
8	C'Sink 6 Grooves	20	S. Drill	1	1	0.40	-	2.5
9	File Burrs		Bench	1	1	0.40	-	-
10	Roll Teeth	50	Lathe	1	1	0.25	0.3	-
	Final Inspection before Hard'							
11	Harden - Carbonize - Brush							
	Reheat - Quench - Draw & Wash							
12	Sand Blast - Polish for							
	Brinell Test - Test with File							
13	Grind Bore	22	Heald	1	1	2.4	-	0.5
	Final Inspection							
14	Repairs							

FIG. 2 TYPICAL OPERATION SHEET

of different capacity, each man has a different problem. An automobile transmission was used as the basis of this problem as it was not too complicated and was a mechanism with which practically all students are familiar. Experience has demonstrated it to be a wise choice. One of these transmissions (Fig. 1) was mounted in the drawing room so that all students could acquaint themselves with its design and operation. Each student is supplied with sheets detailing the operations on each part. A typical operation sheet is shown in Fig. 2. Also, as an assistance toward a thorough understanding of the problem, manufacturers' catalogs of all machines used were obtained and made available, each machine number as given on an operation sheet referring to a definite machine. For convenient reference each kind of machine used was given such a number, and data of which the following are typical were supplied for each machine.

MACHINE NO. 1

Double-Spindle Hartness Flat Turret Lathe, 3 in. by 36 in.
Mfd. by Jones & Lamson Machine Co., Springfield, Vt.
Floor Space, 4 ft. by 10 ft.
Rated Horsepower, 7.5.
First Cost, $3800.

While the data by which a student determines the number of machines required in his plant are based on one type of transmission, his plant is laid out as it would be for the manufacture of several types. That is, he does not lay out what would be practically a continuous industry.

The first step in the problem is to determine the number of machines of each kind required. This is accomplished by filling out a chart as shown in Fig. 3. From the operation times as given on the operation sheets the number of hours per month required for each machine, for each part, can be computed, and a summary of vertical columns gives the total number of machine hours per month required of each machine. The shop is assumed to run 210 hours per month. If the machines are assumed to operate on the average 80 per cent of this time, then the average number of hours that a machine will operate per month is 168. Dividing the total of a column by 168 gives the number of machines of that type required. As was previously pointed out, each student is assigned a different number of transmissions per month as the capacity of his shop. For this reason the number of machines determined by each student will vary and in this way each will be working on a different problem.

Having determined the number of machines required, the student draws floor plans for a factory to house the machines. Because it makes a more comprehensive problem, a multi-story type of building is required, constructed in conformity to the requirements of the Industrial Code and Labor Law as to exits, stairways, sanitary conveniences, etc. The machines are located for convenient flow of material and also grouped to assist in proper supervision. The chart shown in Fig. 3 is at this time of considerable value as it shows at a glance where and for what purpose a machine is used.

ORGANIZATION AND PRODUCTION CONTROL

Having the floor plans and data regarding machines, it is possible for the student to attack a large variety of problems. The first one attempted is to make an organization chart for the management of the shop. The theory of organization has been previously discussed in lectures, and the development of functional (or line and staff) organization, as industries grew too large for line organization, has been explained. In the discussion of organization and organization charts the charts shown in Principles of Industrial Organization, by D. S. Kimball, and charts like those by L. V. Estes which have appeared in *Industrial Management* are very helpful. The student is examined on his knowledge of the duties and functions of each department shown in his chart to make sure he has studied the subject and expended thought upon his work. It is considered far more essential that a student expend thought on his solution of this problem than that he arrive at an ideal solution.

Mechanisms of control and the use of orders and returns are next studied. It is assumed that a customer's order is received calling for some parts which can be shipped from stock and some which have to be manufactured. This originates shipping orders

PART NO	PART NAME	NO. PER TRANSMISSION	MACHINE	1	2	3	4	5	6	7	8
				DOUBLE SPINDLE HARTNESS FLAT TURRET LATHE 3" x 36" J & L MACHINE CO.	14" SURFACE GRINDER PRATT & WHITNEY	NO 3 B. LAPOINTE BROACH J.M. LAPOINTE CO	SINGLE SPINDLE HARTNESS FLAT TURRET LATHE 24" x 24"	14" FAY AUTOMATIC LATHE J & L MACHINE CO	NO 6 SEMI-AUTOMATIC POTTER & JOHNSON	NO 14 NATCO MULTIPLE HAT AUTO TOOL CO.	NO 53 ACME AUTOMATIC LATHE HAT ACME CO
1	Countershaft Drive Gear	1	2425	210	525	31.25		42.4			
2	Large Sliding Gear	1	2425			75.70			47.7		
3	Idler Gear	1	2425	29.9		75.70					
4	Small Sliding Gear	1	2425			75.70			59.0	145.4	
5	Interlock	1	4850								24.13
6	Cover	1	2425								
7	Screws for Cover, Rear Bearing Cap & Brake Lever Bracket	10	97000								227.8
8	Cover Cap	1	4850				115.2				
9	Rear Bearing Cap	1	2425	230.5	285						
10	Front Bearing Cap	1	2425	143.5	365						
51	Pin for Part No 43	1	9700								13.13
52	Collar for No 43	1	4850								
65	Lock Wire for No 64	2	9700								
60	Brake Pawl Rod-Upper	1	4850								
	Total Hours for 4850 Transmissions per Month			464.9	1618	5510	780	1432	1075	1454	5305
	Number of Machines for 4850 Transmissions per Month			3	2	4	1	15	7	9	4

FIG. 3 CHART USED IN DETERMINING THE NUMBER OF MACHINES REQUIRED OF EACH KIND

production orders, work orders, material issues, etc. It has been found that the use of charts similar to that of Fig. 4 helps to fix the relation of these various forms in the student's mind.

It is pointed out that a separate work order is issued for each operation on each part or lot of similar parts. The method of handling the issuing of stores and keeping of a balance of stores ledger is explained and the forms usually met with are discussed in detail. The student then devises sample forms such as the following: A requisition, material issue, work order, shipping order, production order, and job time card. He must also be able to trace the path of each one of these forms in its travel through the shop, telling who fills in each item and his authority or reason for doing so; also what information is obtained from each form, by whom, and what uses he makes of it. It has been found that this problem gives the student a very real knowledge of the method of using orders and returns for the control of production; he also

gains a knowledge of good practice in the make up of forms and an opportunity is presented for interesting him in the subject of modern job-time recorders, cost-keeping methods, etc.

COST ANALYSIS

After several lecturers on the subject of cost finding and demonstration of the fact that accurate costs depend on accurate distribution of expense, the various methods in use for this distribution are explained and the student computes an hourly expense charge by the machine-rate method for each machine in his plant. To determine these rates, the student compiles a chart such as that shown in Fig. 5. Such items as cost of pulleys, shafting, motors, tools, jigs and fixtures, furniture, miscellaneous equipment and supplies, and indirect labor have to be assumed and are given to each student on the basis of the number of transmissions he is to manufacture each month. This problem also presents the opportunity for the student to familiarize himself with the operation of a modern duplex adding machine which he uses for totaling columns, etc., in this problem. During this problem the student also gains a speaking acquaintance with the subject of depreciation, as items of depreciation enter into his computations and he attends a number of lectures dealing with the various phases of this subject. The advantages of a good cost-keeping system are explained, as well as the advantages and also disadvantages of the machine-rate method for distribution of expense.

After studying the subjects of routing, scheduling, and dispatching the student is in a position to appreciate the advantages of the Gantt chart which has been so ably explained by Wallace Clark in his recently published book. Using the standard Gantt chart forms each student makes out man-record charts for several operators assumed to be employed in his plant and covering a period of one month for each. He makes all his own assumptions as to the amount of scheduled work each man does, failure to achieve schedule, etc. Next he makes a machine-record chart for the machines operated by the workmen he has listed in his man-record chart, and the two charts must correctly correspond. Finally he makes a layout chart for these same machines and this chart must correspond in each detail with the other two charts. When a student has demonstrated that he can make out these three charts all correctly correlated, it is fairly certain that he has gained a working knowledge of the use of the Gantt chart.

These are the problems now used in this course. It can be readily seen that many others could be devised. The outstanding advantages of the course are that the student has definite, specific problems to attack which are absolutely general in their application. The knowledge he gains makes a lasting impression, and from the standpoint of instruction his work can be easily and accurately supervised, for while each student's problem differs from that of his neighbor, still they are all based on the same data and the same rules will apply to all.

It might be well to mention the fact that this is a course given to men who are students of mechanical engineering and receive the M.E. degree. All students who are to receive this degree have a common curriculum until the end of the junior year, which includes courses in elementary economics and English. During their junior year they all attend a course of lectures on industrial organization which is elementary in scope. In the senior year those who wish to specialize in industrial engineering take the course outlined in this article together with the lectures which it accompanies and a lecture course in safety and fire protection, and may elect courses in accounting, business law, industrial relations, and related subjects.

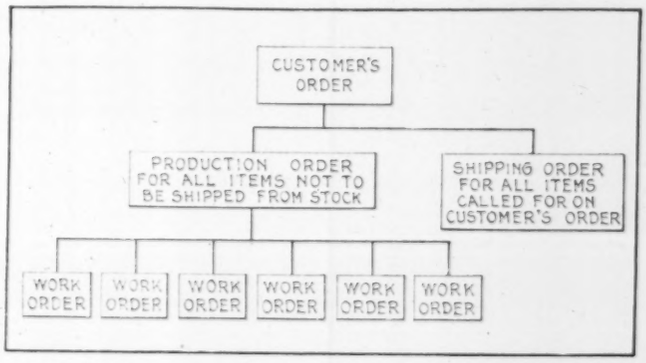

FIG. 4 CHART SHOWING DISPOSITION OF CUSTOMER'S ORDER

In teaching this subject the lack of standardization of industrial terminology is an ever-present cross. Even the correct name for the subject itself is a matter of personal taste. Here it is called "Industrial Engineering." Another educator might call an identical course "Industrial Management," "Factory Management" or any of a score of titles. In one plant a certain form is called a "Material Issue," in another the same thing is a "Requisition." The first plant keeps an article in a "Store Room," the second keeps the same article in its "Stock Room." Standardization of nomenclature is certainly a goal worth the effort to achieve, and may the committee appointed for that purpose have all manner of success.

1	2	3	4	5	6	7	8	9	10	11	12	13	14	15	16	17	18	19	20	21	22
MACHINE NO	MACHINE	NUMBER OF MACHINES	AREA OF ONE MACHINE	PRESENT VALUE	SCRAP VALUE	LIFE IN YEARS	TOTAL AREA OF GROUP	FLOOR AREA CHARGEABLE TO GROUP	HOURS USED PER MONTH	HORSEPOWER RATING	HORSEPOWER-HOURS	POWER COST FOR GROUP	DEPRECIATION FOR GROUP	HOUR CHARGE FOR GROUP	SPACE CHARGE FOR GROUP	VALUE CHARGE FOR GROUP	TOTAL CHARGES FOR ONE MONTH	HOURLY RATE FOR GROUP	HOURLY RATE	ROUNDED RATE	MACHINE NO
1	Double Spindle Turret	6	40	3800	1410	15	240	1281	1008	75	3780	37.00	121.90	30.96	83.54	4.38	711.40	4.23	.705	.71	1
2	14" Surface Grinder	5	89.8	2400	890	15	449.4	2398	840	15	6290	61.60	65	25.80	156.47	2.30.50	539.37	3.213	.643	.65	2
3	Lapointe Broach	4	35.4	800	266	12	141.6	756	672	8	2680	26.24	22.90	20.60	49.42	61.50	180.66	1.075	.269	.27	3
4	Single Spindle Hartness	11	40	3000	1130	15	440	2346	1185	5	2960	28.99	177.50	36.40	153.02	6.34	1029.91	6.13	.557	.56	4
5	14 Fay Automatic Lathe	15	37.2	2500	928	15	558	2981	2520	7.5	9450	92.49	203	77.30	194.52	7.20	1287.31	7.66	.510	.52	5
21	Fellows Gear Shaper	41	20.7	2675	802	10	849	4521	6880	3	10320	101.19	1050	203.20	295.02	2110	3759.41	22.39	546	.55	21
31	Trans Case Boring Mach	10	47.25	3000	900	10	472.5	2521	1080	10	9440	82.29	287.50	51.20	164.52	5.77	1162.51	6.94	694	.70	31
44	Universal Milling Machine	9	55	1650	612	15	495	2641	1512	5	3780	36.99	80.50	46.40	172.72	285.50	622.11	3.70	.411	.42	44
53	"Lo-Swing" Lathe	14	29.8	1440	635	15	417	2221	2352	10	11760	115.09	109.20	72.20	145.02	3.87	628.51	4.94	.353	.36	53
71	Butt Welder	2	15	400	120	10	30	160	336	10	1680	16.49	7.66	10.30	10.45	15.38	60.28	.359	.179	.18	71
72	Emery Wheel	2	4.88	150	50	12	9.76	52.1	336	3	504	4.93	2.16	10.30	3.40	5.71	26.56	.158	.079	.08	72
74	Grinding Wheel	2	1.56	140	46.50	12	3.12	16.6	336	2	336	3.28	2	10.30	1.08	5.38	22.04	.131	.065	.07	74
	Bench	17	16	50	5	15	272	1450	2790	-	-	-	4.58	85.40	94.50	16.32	200.80	1.195	.071	.08	B
												1283.00	398.36	1500.00	3228.65	1085.50	20,943.51				

FIG. 5 TYPE OF CHART USED IN DETERMINING MACHINE RATES OF THE VARIOUS MACHINES IN A PLANT

Discussion

J. O. Keller[1] submitted a written discussion in which he called attention to the fact that Prof. Hugo Diemer had early recognized

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the importance of the propositions set forth in the paper, and as a consequence the course in industrial engineering at Pennsylvania State College, under his able direction, had comprised laboratory courses in time-study work, job routing, factory planning, etc.; and now included such courses in cost accounting, cost analysis, and production control.

His criticism of the course described by the author was that it was largely "cut and dried" and that not enough was left to the initiative of the students, most of his work consisting in the filling in of forms. The course, as he understood it, however, was somewhat limited in regard to time, and to cover so large a field rapidly and effectively some such procedure had probably been found necessary.

George H. Shepard[2] outlined the course in industrial engineering at Purdue University and pointed out that although developed independently, both it and the course at Cornell had evolved along the same lines. In the matter of time study he considered the setting of standard time only one of the objects to be sought. It was of at least equal importance to determine a correct method and to draw up a standard-practice instruction for it. In order that the engineer might be able to do this, it was essential that he should be able to make his own job analysis, and at Purdue they especially avoided giving to the student mimeographed sheets which enumerated the elements of each operation.

In all the work at Purdue emphasis was placed on principles. The characteristic methods of mass production were worse than useless to the jobbing and repair plant, and neither of them employed methods that could be applied to a department store. Yet department stores were beginning to seek industrially educated engineers to serve as the key men of their organizations. Management problems were infinite in their variety, and it was only by doing his own thinking on the basis of general principles that the manager could deal with them.

P. F. Walker,[3] who opened the oral discussion, said that the student training outlined in the paper was identical in many respects with work that was being developed under his supervision at the University of Kansas. It differed most in that in the latter institution an attempt was made to carry the student through the various commercial and business bases of the development of an industry. The points stressed were (1) the market and a sales policy for its development; (2) considerations affecting location of the plant in both general and local aspects; (3) the financial analysis of manufacturing enterprises of various types but with emphasis on machine building, on which the design was to be based. The aim was to plan an industry as to both its commercial and its physical aspects. He agreed with the author as to the value of such courses, for without experience in the actual laying out of plants and in making cost analyses the teaching of plant organization and production control lacked definiteness.

H. Wade Hibbard[4] said that he had three criticisms to offer. The first, which had already been mentioned by others, was on the use of mimeographed sheets which enumerated the elements of each operation to be studied and the time when each reading was to be taken. He believed that a student learned very largely by his mistakes and that everything should not be cut and dried for him. It was the bane of many a large engineering school that so much work of this kind had to be done. His second criticism was as to the use of lantern slides, which gave merely fleeting impressions that were forgotten by the student as soon as they passed off the screen. His third criticism was that the automobile transmission which was used as the basis of the student's mass-production problem was too complicated, and that the principles involved could be better taught by making use of examples of a simpler nature.

J. A. Shepard,[5] speaking from the viewpoint of a manufacturer, commended the cut-and-dried method because, if well devised, it seemed to him to be the most effective way of teaching the best methods to those at the beginning unfamiliar with manufacturing procedure. He stressed the importance of including in such instruction some consideration of the material-handling problems that entered in between the successive processes of production, in order that the student might not fail to provide for material-handling equipment in his building layouts.

F. B. Gilbreth[6] deprecated the use of the cut-and-dried method for the reason that it was wrong to teach students to repeat the mistakes of their predecessors. The time study that had been spoken of was a rule-of-thumb procedure because it determined how long it took to do work but did not determine the method. Time study could be eliminated so far as recording the facts was concerned by using a motion-picture camera for that purpose. Accuracy of time, to his mind, was of secondary importance as compared with accuracy of method. His firm had a very complete collection of films of operations performed by the best demonstration procurable, and these they would be glad to lend to educational institutions for instructional purposes.

John J. Long[7] thought that the cut-and-dried method of teaching could be defended on the ground that it brought to the attention of the untrained student's mind the best practice the instructor knew. Many of the problems of industry were not those of production alone, and he would like to see greater stress placed on the financial and administrative side of industry, particularly with reference to the small shop, for most American shops were those with a personnel of 100 or less.

Dexter S. Kimball[8] said that the thing that was cut-and-dried today might be a totally different thing tomorrow. For many years students at Cornell had been instructed in testing materials in the old way, each one making his own mistakes and repeating those of his predecessors. Then it was decided to make a change and an instructor was accordingly employed who was skilled in the use of the machine and who was able to demonstrate to the students just how a test should be made and how it should be recorded. This had been their practice for a number of years past and, so far as he knew, there were none who wanted to return to the old methods.

The author, in closing, said that he believed any engineer would admit that mass production as it existed today was not essentially a bad thing, and that no one would care to revert to the old handicraft civilization. It was a fact, however, that the problems of mass production were now involved in the instruction of large bodies of students, and he did not believe that the so-called cut-and-dried method was inherently bad. It had been his experience that a student left to himself had an infinite capacity for wandering around and not getting anywhere, and no matter how carefully cut and dried a problem might be, he would make enough of the mistakes that some thought were necessary for him to make, in order to succeed as a student.

He was obliged to disagree with Professor Hibbard in regard to the complexity of the basis used in the mass-production problem. An automobile transmission was about the simplest mechanism that could be selected to use in teaching students real manufacturing methods, and in addition it was one with which students were fairly well acquainted. Further, he believed in using lantern slides, for when addressing students he felt it was much easier to make a lasting impression on their minds through the eye rather than through the ear.

He would not attempt to uphold the time-study work of the course in all its features, but it was proper to say that many to whom it had been given in the past had come back and told of the great benefit they had derived from it.

In regard to material handling, no definite laboratory or drawing-room work had been included in the course, largely because of time limitations. However, an endeavor was made to bring out the fact that the layout of a plant was governed largely by the sort of material-handling equipment required in the particular industry for which the plant was to be built.

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[3] Dean, School of Engineering, University of Kansas, Lawrence, Kan. Mem. A.S.M.E.

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[5] Vice-president and chief engineer, Shepard Electric Crane & Hoist Co., Montour Falls, N. Y. Mem. A.S.M.E.

[6] Frank B. Gilbreth (Inc.), consulting engineers, Montclair, N. J. Mem. A.S.M.E.

[7] Industrial engineer, Schapiro & Schapiro, New York, N. Y. Assoc. Mem. A.S.M.E.

[8] Dean, College of Engineering, Cornell University, Ithaca, N. Y. Past-Pres. A.S.M.E.

The Thermal Conductivity of Liquids

Details Regarding Apparatus Employed in a Recent Investigation of the Subject—Difficulties Encountered in Making Tests—Radiation Corrections—Results Obtained and Conclusions Reached

By F. L. KALLAM,[1] LONG BEACH, CAL.

ALTHOUGH the thermal conductivity of liquids has been for a long time a favorite field of research for the physicist, it is still in a very undeveloped and unsatisfactory state. The subject is of profound importance in dealing with problems of heat engineering, and while the present investigation has not progressed far enough to yield definite conclusions, it has nevertheless resulted in the devising of apparatus which is believed to be suitable for obtaining the conductivity of liquids within a reasonable degree of accuracy.

In investigating the conductivity of liquids nothing more than relative values have been obtained, and even the method of arriving at results is often obscure. As early as 1800 Count Rumford showed the low conductivity of water by heating a tube at the middle of its length, boiling the water at the top by convection currents and leaving a chunk of ice at the bottom unmelted.

Despretz in 1838 made a series of experiments with a tin vessel full of hot oil kept at constant temperature floating on the surface of water in a large container, and a series of thermometers one below the other in a containing vessel. In this manner he found that the conductivity of heat in liquids obeys the same laws as in solids, but is much less pronounced. Bottomley in 1881, using a modification of Despretz' method, poured hot water on the top of a mass of water nearly filling a cylindrical wooden tank. Readings were taken from time to time of two thermometers one a little lower than the other, which gave the difference of temperature between the two sides of the intervening stratum, and the quantity of heat conducted in a given time through this stratum was obtained from readings of the temperature of the whole mass of water below on an upright thermometer with an exceedingly long cylindrical bulb extending downward from the center of the stratum in question nearly to the bottom of the vessel. The value for conductivity as determined by this method varied from 5.8 to 6.67 (English units).

Guthrie examined the conductivity of liquids by using two hollow brass cones placed near together, the top of one pointing upward, that of the other downward. The distance separating the bases could be regulated and the liquid to be examined was introduced between them by means of a pipette. The base of the upper cone was kept at a constant temperature by a current of hot water, and the lower cone was converted into an air thermometer by simple methods. By varying the distance between the bases of the cones as well as the liquids introduced between them, and noting depressions of the air thermometer, a measure of the resistance to the passage of heat of different liquids was obtained.

Weber in 1880 used two small copper disks separated by pieces of glass, the space between the disks being filled with liquid; the lower disk was laid on a smooth block of ice whose temperature it rapidly assumed and measurements were made of changes in temperature of the upper disk with a thermoelectrical arrangement. Lees employed the same method with some refinements, but used an ebonite ring to enclose liquid between the disks. On the whole, nothing definitely satisfactory seems to have been developed hitherto in the way of apparatus or experimental methods for the determination of the true conductivities of liquids.

Mathematical Expression for Thermal Conductivity

When convection is eliminated, the formula embodying the coefficient of thermal conductivity may be deduced in the same manner as for solids.

If Q = B.t.u. flowing per square foot per hour

$T_1 - T_2$ = temperature drop in deg. fahr. between two planes cutting the liquid parallel to each other, and perpendicular to the direction of flow

A = area of section through which heat flows, sq. ft.

S = time of flow in hours

L = depth of liquid or perpendicular distance between two planes, in.

k = conductivity of the liquid in B.t.u. per sq. ft. per in. per deg. fahr. per hour

then

$$Q = \frac{k(T_1 - T_2)AS}{L} \quad \text{or} \quad k = \frac{QL}{(T_1 - T_2)AS}.$$

If it is assumed that k is a straight-line function of temperature, the mean conductivity between any two points is the arithmetical mean of the conductivities at these temperatures. But throughout this investigation K the mean conductivity is taken as k for the sake of simplicity.

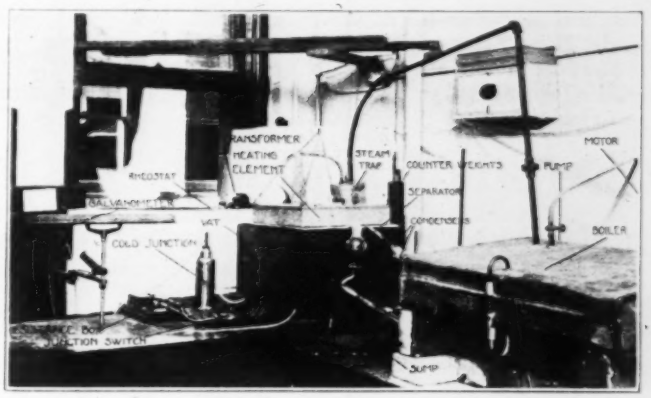

Fig. 1 General View of Apparatus Used by Author in Investigating the Thermal Conductivity of Liquids

Principles of the Apparatus

In studying thermal conductivity, convection currents must be eliminated. Therefore it is quite certain that the liquid to be tested must be heated at the top.

The material of the container must have a lower conductivity than that of the liquid to be tested or heat will be transmitted through the walls of the vessel faster than through the liquid, and this will lead to convection currents. Hence a material such as wood must be chosen for the container.

The quantity of heat transmitted to the liquid must be known and this would be readily obtainable by supplying the heat from steam at atmospheric pressure and measuring the amount of steam condensed. Hence the use of a steam heater follows. To avoid the complication of radiation losses, the heater must be jacketed with steam at the same pressure and temperature except where it is in contact with the liquid to be tested, so that all the heat released by condensation will pass into this liquid.

Temperatures must be known at different strata instantaneously, and with the strata close together thermometers are eliminated not only on account of lag but because of mechanical difficulties.

The apparatus is based on these principles and the method employed may be briefly described as follows:

The liquid to be tested is placed in a vat, which is itself a poor conductor and which contains thermocouples at known strata.

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Abstract of a thesis submitted by the author in June, 1922, to the Department of Mechanical Engineering and Committee of Graduate Study of Leland Stanford Jr. University in partial fulfillment of the requirements for the degree of Engineer. Awarded A.S.M.E. Junior Prize for best paper during 1922.

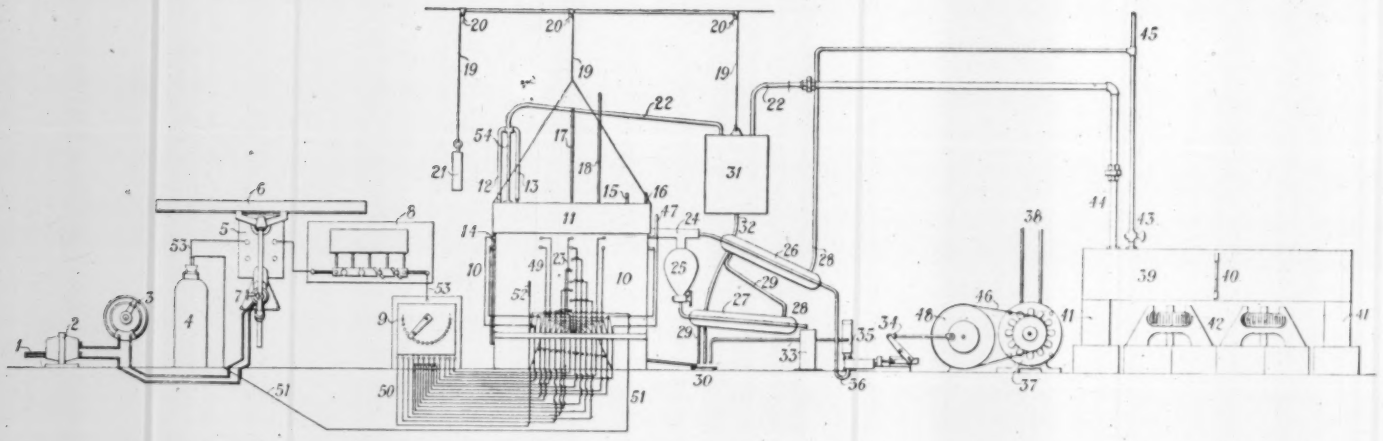

FIG. 2 DIAGRAM SHOWING ARRANGEMENT OF AUTHOR'S APPARATUS FOR DETERMINING THE THERMAL CONDUCTIVITY OF LIQUIDS

1 110-volt a.c. main
2 Bell transformer
3 Rheostat
4 Thermos bottle—cold junction
5 Paul galvanometer
6 Galvanometer scale
7 Galvanometer light
8 Resistance box
9 Junction switch
10 Vat, jacket removed
11 Heating element
12 Jacket intake
13 Bath intake
14 Jacket drain
15 Bath exhaust
16 Jacket exhaust
17 Bath thermometer
18 Jacket thermometer
19 Elevating cord
20 Pulley
21 Counterbalance
22 Flexible steam line
23 Thermocouples
24 Bath drain—magnesia covered
25 Steam separator
26 Steam condenser
27 Condensate cooler
28 Intake cooling water
29 Discharge cooling water
30 Sump
31 Steam separator
32 Steam - separator drain
33 Condensate graduate
34 Pump
35 Pump discharge
36 Pump suction
37 Single-phase motor
38 110-volt a.c. main
39 Boiler
40 Water glass
41 Brick supports
42 Gas burners
43 Feedwater
44 Steam main
45 Water supply
46 Driving belt
47 Level glass
48 Driving wheel
49 Copper and iron-constantan leads from hot junctions
50 Constantan leads
51 Copper neutral lead
52 Free thermocouple No. 11
53 Neutral constantan lead
54 Support

Heat is supplied to the surface of the liquid by means of a steam-jacketed bath, the vapor being at atmospheric pressure. The amount of heat liberated is determined from the condensate collected. Hence, knowing the vat dimensions, the temperatures at different strata, strata depth, and heat generated, all for the same period of time, the thermal conductivity is readily obtained from the formula

$$K = \frac{QL}{(T - T_2)AS}$$

DESCRIPTION OF THE APPARATUS

The apparatus is illustrated in the views and diagrams of Figs. 1 to 4, inclusive.

Temperature Measurement. Avoiding thermometers of special and complicated construction, thermocouples of copper-constantan were employed, the junctions being welded into the form of a bead. To support these junctions at the proper locations within the vat, the couples were run through a $^1/_8$-in. glass tube, care being exercised to make a good seal between the wire and the glass. A direct-reading galvanometer of the R. W. Paul reflecting type was found to be a simple and effective measuring instrument. The thermocouples were arranged in the center at intervals of approximately 1 in. from top to bottom of the 7 in. depth of the vat. At the point of contact of heater and liquid five couples were used, and the same number at 1 in. lower depth. From this point to the bottom only one couple at each level was used. A seventeenth couple, not fixed as to location, was used to explore temperatures of the container. As it was impossible to place the junctions exactly one inch apart and it was necessary to measure accurately the actual distances, a single cold junction was provided for the seventeen couples and suitable connections were made so that each couple could be put in series with the galvanometer at will.

Calibration was made in place, with the cold junction kept at 32 deg. fahr. and the water in the vat heated by means of two electric heaters. To make the temperature in the vat uniform the water was agitated with a paddle. Throughout the investigation the cold junction was kept at 32 deg. fahr. by the use of cracked ice within a thermos bottle.

The Vat. The vat containing the liquid to be tested is constructed of $1^1/_4$-in. redwood, with a cross-section of exactly one square foot for facility in calculation. Holes are provided for the insertion of thermocouples at vertical intervals of 1 in. The bottom tapers to a drain plug at the center.

Calked joints and waterproof coatings are provided to secure perfect watertightness. A small glass tube is used to see that the water in the vat is always at the proper level. It was thought that the vat was suitably constructed to prevent excessive radiation losses, but it was ultimately found necessary to enclose it in a radiation jacket, in the form of a slightly larger second wooden box so as to provide a dead-air space around the vat proper.

The Heating Element. As shown in the figures, the heating element consists of two parts, the heater or bath proper and a jacket. The heater is exactly 1 ft. square so that it fits snugly into the vat. The entire heater and the surface in contact with the water is of copper, used because of its high conducting power. The heating surface is not exactly flat but is somewhat dished outward, for the purpose of forcing out any air which might otherwise be trapped between the vat and the heater when the latter is first placed in contact with the liquid. The dished bottom also serves as a sump for all steam condensed. A $^1/_8$-in. brass tube having a right-angle turn is so placed within the heater that it dips down to the bottom of the sump. The end of this tube is flared, with four notches cut in its edge, thus affording what might be termed "feet."

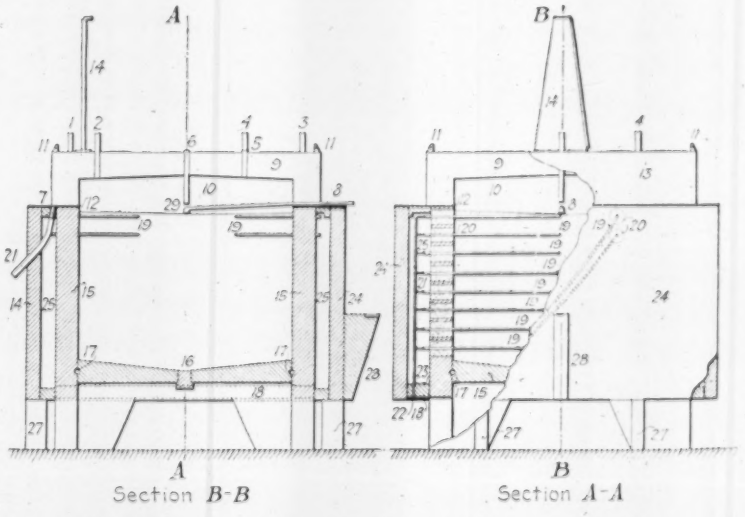

FIG. 3 DETAILS OF HEATING ELEMENT, VAT, AND JACKET

1 Intake to heater jacket
2 Intake to steam bath
3 Exhaust from heater jacket
4 Exhaust from steam bath
5 Jacket thermometer cup
6 Steam-bath thermometer cup
7 Jacket drain
8 Steam-bath drain
9 Galvanized-iron jacket
10 Steam bath
11 Clips for supporting bath and jacket
12 Asbestos packing
13 Heating element, jacket, and bath
14 Support for steam line
15 Redwood vat
16 Drain plug
17 Recessed butt joint
18 Thermocouple lead board
19 Glass-encased thermocouples
20 Rubber stoppers
21 Leads from couples to lead board
22 Leads to junction switch
23 Neutral lead to cold junction
24 Redwood radiation jacket
25 Dead air space
26 Drain from heater jacket
27 Jacket supports
28 Support for steam-condensate separator
29 Pump suction

These are soldered to the dished copper bottom, which leaves four openings in the tube. The latter rises 1/4 in. and then runs parallel to the heater bottom, passing out at the side at the same elevation. Thus, by applying suction at this outer end, the condensate within the heater may be removed. The arrangement of the heater can be studied more closely in the figures. It should be noted that the bottom of the heater is free from the jacket, projecting down into the liquid for good contact. The steam line is divided, part going into the heater and part to the heater jacket. A suitable steam separator is provided at the lowest point in the steam line and as close to the heater as possible. In using the apparatus it is necessary to have the heater and jacket fully heated at the start of the test, hence steam cannot be passed through the heater when it is in contact with the liquid to heat the metal up to working temperature. To facilitate such heating, both the trap and heater are suspended by cords over pulleys and attached to the same counterbalancing weight. Thus they may be readily removed from the test position to a location where they may be heated without affecting the liquid to be tested.

The Condensate Separator. In removing the condensate from the heater there is a possibility of some steam being taken along with it. As the condensate proper is a measure of the heat put into the liquid tested, it must not be contaminated with free steam which has been condensed after leaving the heater. For this purpose a separator is placed as close to the heating jacket as possible. This is nothing but a tee made of glass tubing, with the third leg vertical and connecting directly to a glass chemical separating funnel. The entire tee is covered with magnesia. By means of this tee any steam drawn over passes over the third leg and passes on to the condenser and pump, while the condensate proper, being of greater density than the steam, flows into the third leg of the tee and on to the funnel, where it collects. Suction must not be applied too rapidly or else the condensate will skip the third leg and take the steam route. Before collection in a calibrated vessel the condensate is cooled to prevent possible evaporation due to its high temperature.

Manipulation of the Apparatus

In starting a test the heating element is raised and steam passed through the system. This is continued until both the jacket and bath thermometers indicate 212 deg. fahr. Any steam condensed in the heater must be completely removed, and for this purpose an atomizer was found to give the best results. The vat must be completely filled with liquid, so that when heater is lowered in contact with it, some of the liquid will be forced over the top. This insures good surface contact with the heater. With the heater in contact with the liquid, the test starts. The separator must be instantaneously connected to the heater and the two condensers put into operation. Just prior to placing the heater in contact with the liquid, initial readings on the 17 couples, the two heating-element thermometers, the room-temperature thermometer, and the graduated condensate vessel must be taken.

Difficulties in the Tests

A large number of tests were made but only three are recorded in the report of the investigation, the others being discarded because of such difficulties as inability to maintain constant temperature due to lack of steam, leakage at the entrance of the glass thermocouple, tubes, and short-circuiting of the thermocouples. The vat rapidly deteriorates and at the end of a certain time becomes unfit for use. Blisters are formed on its exterior coating due to water which seeps from the interior. This is a sign that the vat is becoming waterlogged and the temperature of the liquid under test ceases to increase because water flows through the sides of the vat in small streams. Hence in using a wooden container some preparation must be applied to the interior which will prevent the liquid from coming in contact with the wood. No suitable material has yet been found, and the only remedy is to dry out the tank for two weeks when it becomes waterlogged. Intermittent working of the suction pump must be avoided, and other minor difficulties are inescapable.

Radiation Corrections

Even when the vat is jacketed, corrections must be applied to cover radiation losses. To secure these corrections the following procedure is employed: The water in the vat is heated to the boiling point and a calibrating lid is placed in position; this lid is merely a cover of the same material as the vat, through which five thermometers project into the water. Temperatures are taken from time to time, and the temperature of the bath is plotted against time. From this curve the rate of cooling at different temperatures of the vat is obtained, measured in degrees of drop per hour. The readings of the thermocouples are appropriately corrected for this drop when computations are made. The corrections are not absolutely accurate, since in determining radiation losses the temperature of the water in the vat is uniform, whereas in the conductivity tests there is a temperature gradient. The correction is of course always added to the readings of the thermocouples.

Computations

The data obtained in the three final tests are elaborately set forth in the report in tables and curves where temperatures at various depths are plotted against time. It is only after three hours of testing that the temperature difference between strata tends toward uniformity. For this reason the temperatures corresponding to

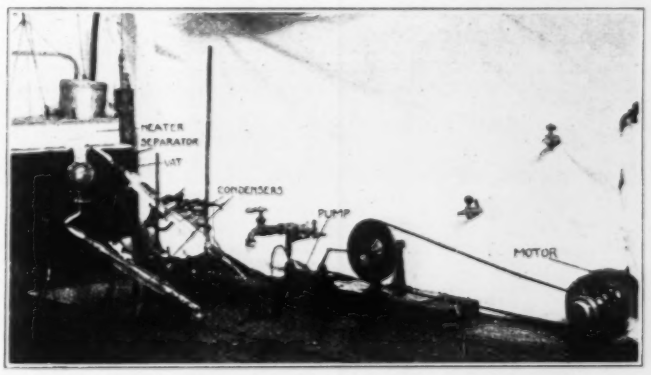

FIG. 4 VIEW OF PART OF APPARATUS WITH BOILER REMOVED, SHOWING PUMP AND MOTOR

3, 4, 5, and 6 hours are selected and arranged in a special table (Table 1) after corrections have been made for radiation losses.

From Table 1, temperature is plotted against the depth for con-

TABLE 1 INSTANTANEOUS TEMPERATURES AT VARIOUS STRATA

(From corrected temperature data of tests Nos. 2 and 3; temperatures in deg. fahr.)

Strata	L, in.	Time, hr.	T_1	T_2	T_1-T_2	Avg. T
B-C	1.25	3	180	130	50	155
		4	186	141	45	163
		5	190	147	43	168
		6	192	152	40	172
B-D	2.06	3	180	107	73	143
		4	187	117	70	152
		5	190	127	63	158
		6	192	130	62	161
B-E	3.19	3	180	85	95	132
		4	186	97	89	141
		5	190	103	87	146
		6	191	108	83	149
B-F	4.13	3	180	80	100	130
		4	186	86	100	136
		5	189	90	99	139
		6	191	94	97	142
B-G	5.01	3	181	76	105	128
		4	186	79	107	132
		5	189	82	107	135
		6	192	86	106	139
B-H	5.95	3	180	77	103	128
		4	186	78	108	132
		5	190	80	110	135
		6	191	82	109	136
C-D	0.81	3	131	107	24	119
		4	140	117	23	128
		5	147	125	22	136
		6	153	131	22	142
C-E	1.94	3	131	88	43	109
		4	140	96	44	118
		5	147	103	44	125
		6	152	109	43	130
C-F	2.88	3	131	80	51	105
		4	139	85	54	112
		5	146	90	56	118
		6	151	96	55	123
C-G	3.76	3	130	77	53	103
		4	140	80	60	110
		5	145	82	63	113
		6	153	87	66	120
C-H	4.70	3	130	77	54	104
		4	140	78	62	109
		5	147	81	66	114
		6	152	83	69	117
D-E	1.13	3	105	88	17	96
		4	117	96	21	106
		5	127	103	24	115
		6	133	109	24	121
D-F	2.07	3	107	80	27	93
		4	118	85	33	101
		5	126	91	35	108
		6	132	96	36	114
D-G	2.95	3	106	77	29	91
		4	117	79	38	98
		5	125	83	42	104
		6	132	86	46	109
D-H	3.87	3	106	76	30	91
		4	117	78	39	97
		5	125	80	45	102
		6	131	82	49	106
E-F	0.94	3	88	81	7	84
		4	96	85	11	90
		5	103	90	13	96
		6	109	96	13	102
E-G	1.82	3	87	75	9	82
		4	96	78	13	89
		5	103	83	12	92
		6	109	91	10	104
E-H	2.76	3	88	76	19	82
		4	97	78	22	82
		5	104	82	22	87
		6	109	83	26	96
F-G	0.86	3	79	77	2	78
		4	84	79	5	81
		5	89	82	7	85
		6	95	86	9	90
F-H	1.80	3	79	63	3	77

NOTE.—*B* is located 0.81 in. below the heater.

stant times as shown in Fig. 5 for the second and third of the three tests accepted by the author. The area under these curves must represent the total heat put into the liquid up to the time in question, since the ordinates represent degrees and the abscissas volumes of water, which can be converted into weight by multiplying by the density corresponding to the average temperature. This obviously gives the heat required to raise a mass of water a certain number of degrees in a given time. Thus the area included between the

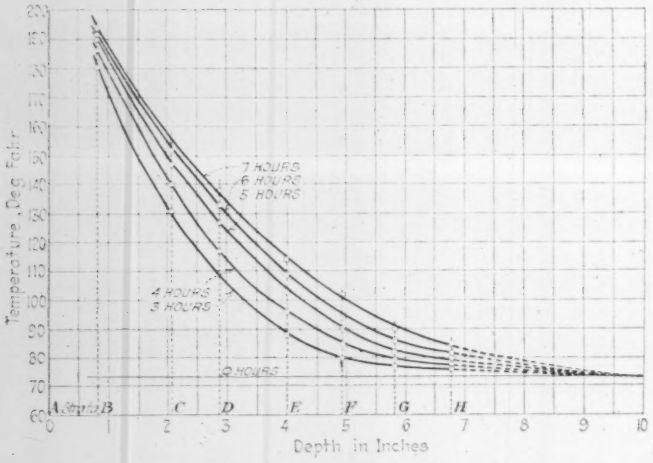

FIG. 5 TOTAL-HEAT CURVES RELATING TEMPERATURE, DEPTH, AND TIME

(Tests Nos. 2 and 3. Computed for a temperature gradient of $(T_1 - T_2)/L = 17$ to 24.)

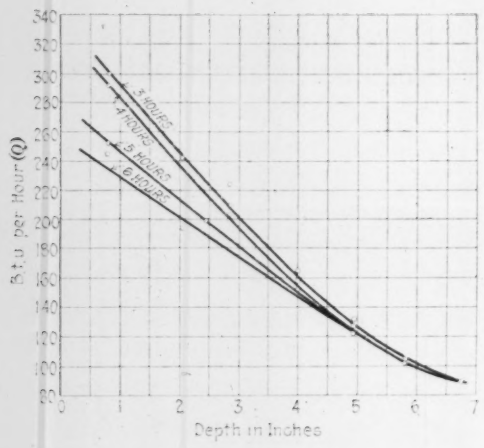

FIG. 6 HEAT-FLOW CURVES DERIVED FROM TOTAL-HEAT CURVES WHERE $(T_1 = T_2)/L = 27$ TO 24

3-hr. and 4-hr. curves is the amount of heat put into the liquid between the third and fourth hours of the test.

In this connection, it will be noticed that the curves of Fig. 5 are extended past their lowest temperature point down to the initial temperature of the water by continuing them smoothly with dotted lines. The author believes that this assumption will yield more nearly correct results than can be obtained by stopping the curves at the lowest temperature reading.

By taking the area between any two time curves starting at any desired depth and planimetering the area for the remaining depths, the heat-transfer rate at the initial depth is found in B.t.u. per hour. This gives Q in the formula direct. Fig. 7 derived in this manner gives the heat-transfer rate at each of the desired strata for the representative tests 2 and 3.

As regards the unit of heat corresponding to a given area in Fig. 5, one inch along the axis of abscissas represents a rise in temperature of 20 deg. fahr. One inch along the axis of ordinates represents 1 in. of depth, which is equivalent to $^1/_{12}$ cu. ft. of water inasmuch as the vat has a surface of 1 sq. ft. From this it follows that 1 sq. in. of area represents 104 B.t.u.

Although some corrections have been made in these curves for radiation losses, a further correction should be made by comparing the rate of heat delivered to the liquid in the vat with the rate of supply from the steam condensed—which may readily be computed from the weight of condensate and the latent heat at the given pressure. The condensate method gives approximately 1.16 times the total heat that the graphical method does and values of conductivity should be multiplied by 1.16 accordingly.

After this work has been performed it is possible to determine the thermal conductivity K from the previously developed expression

$$K = \frac{QL}{(T_1 - T_2)AS}$$

For example, we shall compute K for tests 2 and 3 between strata B and F, the distance between the two being 4.13 in. Also we desire K at the end of 3 hr., at which time $T_1 = 180$ deg., $T_2 = 100$ deg., and Q, the rate from the curves of Fig. 6 for 3 hr., is 300

(Continued on page 511)

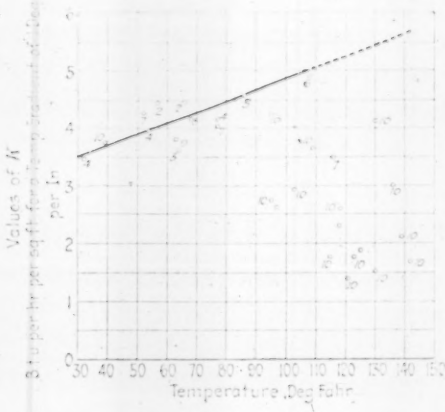

FIG. 7 VARIATION OF THERMAL CONDUCTIVITY OF WATER WITH TEMPERATURE

(Authorities: 1, Wachmuth; 2, Winklemann; 3, Chree; 4, H. F. Weber; 5, Graetz; 6, Lundquist; 7, Lees; 8, Milner and Chattock; 9, R. Weber; 10, Present test.)

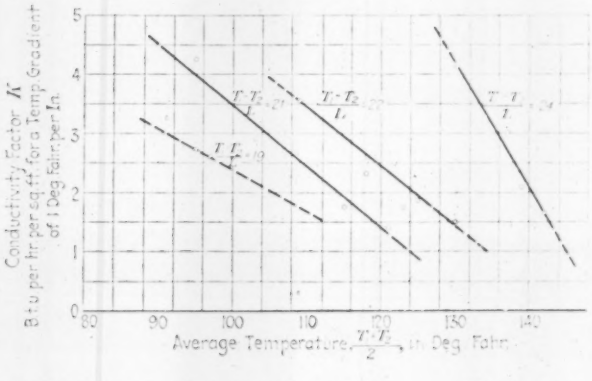

FIG. 8 VARIATION OF THERMAL CONDUCTIVITY OF WATER WITH THE TEMPERATURE GRADIENT

(K found from $QL/(T_1 - T_2)AS$; $T_1 - T_2$ = temperature drop in deg. fahr. between lamina faces; L = perpendicular distance in inches between lamina faces.)

TABLE 2 VALUES OF K COMPUTED FROM TESTS

Test Nos.	Strata	Time S, hr.	T_1	T_2	$T_1 - T_2$	L	$\frac{T_1 - T_2}{L}$	Q	Avg. T	K
2 & 3	B-F	3	180	80	100	4.13	24.2	300	130	4.13
		4	186	86	100	4.13	24.2	292	136	4.01
		5	189	90	99	4.13	24.0	252	139	2.10
		6	191	94	97	4.13	23.6	235	142	1.67
2 & 3	C-E	3	131	88	43	1.94	22.2	242	109	3.64
		4	140	96	44	1.94	22.7	235	118	2.59
		5	147	103	44	1.94	22.7	212	125	1.87
		6	152	109	43	1.94	22.2	200	130	1.50
1	C-F	3	124	67	59	2.88	20.5	265	95	4.23
		4	134	73	61	2.88	21.2	248	103	2.92
2 & 3	D-E	5	127	103	24	1.13	21.2	186	115	1.75
		6	133	109	24	1.13	21.2	178	121	1.40
1	D-E	4	106	84	22	1.13	19.4	214	95	2.74
1	D-F	5	116	79	37	2.07	18.9	235	97	2.63
1	C-G	6	155	82	73	3.76	19.4	270	118	2.32
2 & 3	C-F	6	151	96	55	2.88	19.1	201	123	1.75

Sectionalization and Remote Control of High-Pressure Steam Lines

By PETER PAYNE DEAN,[1] STAMFORD, CONN.

This paper is intended to bring out need of protection other than that of hand-operated valves for high-pressure steam lines in case of failure of pipe or fittings, and deals with the remote operation of various types of valves by electric motor or steam piston in order to quickly and safely stop escaping steam. It is based on a careful study of valve construction and methods of power operation made by the author during the last few years, and on experience gained with a large number of electrically operated valve installations which he has designed and placed in practically all of the important power-generating stations in the country, and gives particulars regarding the construction of various protective devices, valves, and motor-control systems that are available.

STEAM pressures as high as 350 lb. per sq. in. with 700 deg. fahr. total heat are now frequently encountered, and no doubt 500 lb. will soon be standard practice. Under such pressures every piece of valve, piping, and fitting metal will be subjected to an excessive strain, which, when coupled with high temperature, may produce a result akin to metal fatigue, about which we have very little knowledge and even less experience. Further, no safe standard has been universally adopted for 500-lb. steam fittings, bolts, flanges, expansion loops, etc., but this will very probably come after more exhaustive time tests.

The protection of human life, however, should be the first consideration, and after that, continuity of plant operation and protection of property; and surely all will agree that protection beyond that found in the average plant is necessary. We can only make every attempt to prevent failure by careful design, and then provide, as a matter of safety, the best possible safeguard if it does happen. Those who were present at the tests made at the Essex power station of the Public Service Co., Newark, N. J., on January 28, 1922, and watched the spectacle of high-pressure steam calmly blowing to atmosphere at a velocity of over six miles a minute, do not need to draw much upon their imaginations to visualize the damage that such a discharge would do in a boiler or turbine room.

Location of Master Valves

The master valves should be those nearest the boiler, and either the automatic stop checks or gate valves in the boiler lead. Valves at this point protect the complete piping system and would not be affected by mechanical movement of a broken line, which would possibly throw excessive strain on a gate valve, causing it to jam. Automatic stop-check and boiler-lead gate valves are seldom above 10 in. in diameter and without doubt can be closed in from fifteen to thirty seconds, so that any or all of the boilers may be cut out at will.

Unless all the boilers are isolated there would be a reverse flow from the header back through the break, so that a master valve is also necessary in the boiler lead at the header, which in any event is usually included. Closing both the boiler-lead valve and this last-mentioned valve completely isolates one boiler.

Boiler-lead valves are suggested for three very good reasons:

a They protect the whole system of piping from the superheater and are well located for power operation

b If a bad fracture should occur, it would be almost useless to endeavor to locate it and sectionalize the header to stop the flow, on account of the tremendous volume of condensed steam and consequent confusion

c There is no certainty whatever that header valves above 12 or 14 in. in diameter will close under open-end conditions on account of their construction to withstand such strain.

The principal argument against emergency closing off of the boilers is that the plant would be completely shut down. In the author's opinion, if a bad break occurs its location cannot be determined until almost the entire steam supply is stopped. As soon as the break is located, the sectionalizing valves may be brought into play and the boilers quickly brought back on the line.

Even though large header valves can be closed from a distance, the author nevertheless believes that the quick and safe way is to shut off the boilers in sections until the break is stopped. Incidentally, with any remote-closing scheme the boiler safety valves should be piped to a header and blown to atmosphere, as in the event of a boiler shutdown they would all blow together. Some arrangement also of an auxiliary header should be worked out to supply the boiler-feed pumps, as well as a scheme for shutting down the draft fans. Power-operated header valves of large diameter may be depended upon to stop a very bad fitting blow or gasket leak, so for this reason and also for convenience of operation they should be provided with remote control.

To close an 18-in. high-pressure gate valve requires two men

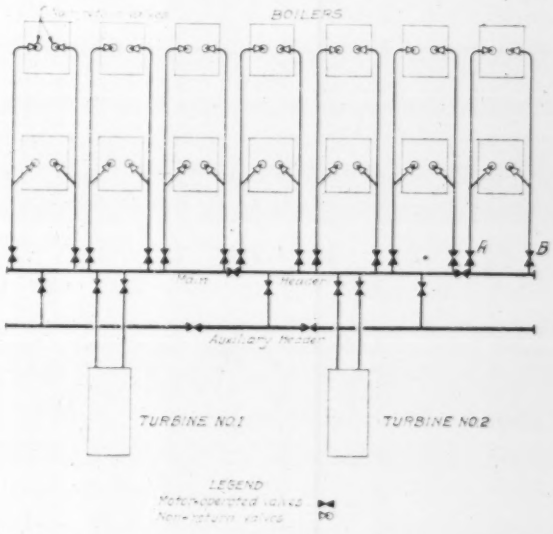

Fig. 1 Boiler and Piping Layout of a Power Station in Which Electrically Operated Valves are Employed
(Colfax Station, Duquesne Light & Power Co., Pittsburgh, Pa.)

for a period of at least fifteen minutes, including transit to and from the valve, not to mention the physical effort necessary and the danger of penetrating a steam-laden alley with very uncertain conditions under foot. Great care is taken to see that main-line valves are tested for tightness and that they are strong and rugged, presumably for assurance that they will not fail during operation or when an emergency arises. Who knows, however, whether they can be even reached in case of trouble? Therefore, why not turn them into real protective devices by providing remote control? It should not take much of an argument to convince those in charge of expenditures that is worth the money.

Fig. 1 shows a typical existing boiler and piping layout in which electrically operated valves are employed. Complete isolation of one boiler from the line may be made by closing any pair of valves *A*, *B*, either singly or from one switch, but this of course leaves the boiler-lead lines unprotected. Further protection may be accomplished by electrically operating the non-return valves shown and closing them together with valves *A*, *B*, in order to prevent reverse flow from the header.

Fig. 2 shows a similar system wherein the boiler-lead valves may be closed separately and the main and branch-header valves used for sectionalizing.

[1] Payne Dean Ltd. Mem. A.S.M.E.

Contributed by the Power Division and presented at the Spring Meeting, Montreal, Canada, May 28 to 31, 1923, of The American Society of Mechanical Engineers. Abridged. All papers are subject to revision.

REMOTE-CONTROL LOCATION

Since any remote-control system must of necessity be electrical, the design and placing of a central switchboard must be carefully thought out. The location should be as near the main means of exit as possible as plant operators in case of trouble will move toward this point. With the open air in view, a man is more apt to regain his composure and presence of mind than when called upon to act hurriedly and then bolt for his life.

Such stations should never be placed in steamproof vaults adjacent to the boiler room and without means of safe exit. They will be useless in emergencies as the operators will not travel in that direction.

The operation of switches to shut the valves should be as simple

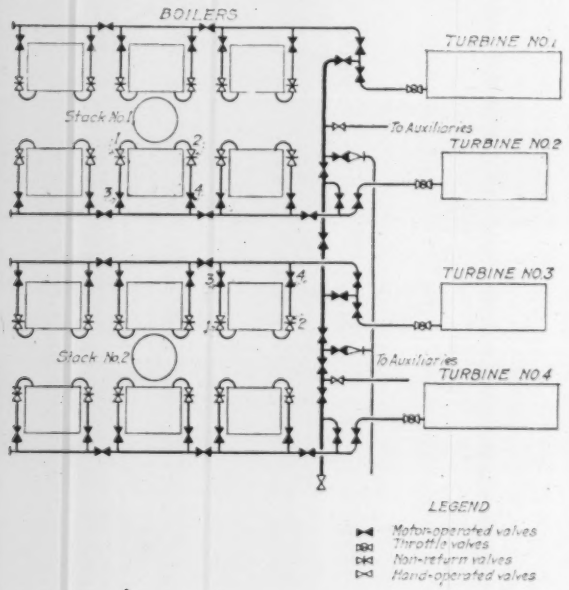

FIG. 2 BOILER AND PIPING LAYOUT IN WHICH THE BOILER-LEAD VALVES MAY BE CLOSED SEPARATELY AND THE HEADER VALVES USED FOR SECTIONALIZING

(Hell Gate Station, United Electric Light & Power Co., New York.)

as possible, preferably by breaking a protective glass window and pushing a button.

Under no consideration is it safe to open master valves from a remote point which may be opened locally, either by hand or electrically. In order to allow the operator to see exactly what he is doing, the face of the valve-control switchboard should be provided with a plainly marked miniature representation of the complete station piping, with push buttons to represent the valves. Red and green indicating lights should be provided to show whether the valves respond and to show the general position of the valves in the complete system.

TYPES OF MASTER VALVES

There are two types of quick-acting valves, the straight triple-duty, and the piston-operated trip valve, both of which are of the globe type. The former is a combination of non-return, stop and "automatic closing on drop of line pressure," performing therefore the three duties. Engineers have not taken kindly to line protection upon drop of pressure; therefore the third duty is operated by an electrically controlled pilot valve admitting steam on top of the closing cylinder. Such a valve and its auxiliary pilot are shown in Fig. 3. This valve depends upon the excess-area piston forcing the disk to its seat against the pressure.

The straight trip valve closes with the pressure and is remotely operated by a solenoid magnetic latch. It is used mostly as an additional emergency turbine stop upon overspeeding, being placed in the turbine lead between the header and throttle trip valve. On account of its somewhat complicated construction, it is hardly suitable for line protection. The triple-duty remotely controlled valve would seem to be the logical type for quick closing, provided that it can be made to close against the extreme velocity which would result with an open-end line blowing to atmosphere and would possibly reach 50,000 ft. per min., depending upon the friction loss in the pipe between the header and the boiler. The action in this case would not be that of a non-return automatic check valve which actually closes due to the reverse flow, but the disk would be forced on to its seat against an extreme velocity in the opposite direction and the action would be somewhat akin to that of stopping a high-pressure nozzle with one's finger. Valves of this description are generally closed by pressure being admitted to the top of the excess-area piston, but test data on closing under extreme velocity conditions do not seem to be available, and until they are, much cannot be said.

A high steam discharge is very apt to carry over slugs of water, and there may be some doubt as to the advisability of instantly closing a valve under this condition. It would seem that a closing time of ten to fifteen seconds would be safer.

MOTOR-OPERATED NON RETURN VALVES

Electrical operation of non-return valves was first suggested by the author and has been adopted by many leading engineers. The operation is the same as that required on the triple-duty design, with the exception that the valve stem is driven by a motor provided with reduction gearing, as shown in Fig. 4. Their action under most extreme conditions is positive, provided that the motor control is of the proper design and application. The valves are ideally located with respect to line protection, being directly at the superheater outlet and where vibration due to the whip of a broken boiler lead would not be excessive on account of the valve being securely anchored. They close easily under normal conditions, and for emergency closing should be provided with ball thrust bearings to eliminate the closing friction as much as possible. The control unit should be bolted directly to the yoke and geared to the stem through the yoke nut, which is of the rising-stem type. The travel of the disk is not more than one-quarter of its diameter, so that quick closing is possible. Although ruggedly built this valve will not stand much jamming, and therefore calls for a special type of motor control to prevent it.

There is only one safe point of closing, and that is when the disks and seats are in perfect contact; to effect this the motor drive should have an automatically operated mechanical clutch completely disconnecting the high-speed armature and gears from the motor at the moment of closing, thus eliminating its momentum and allowing the motor armature to freely come to rest. Without this clutch the limit switch must be set to stop the motor before the valve reaches its seat, and a certain amount of drift due to the momentum of the armature and gears must be depended upon to seat the valve.

This drift depends upon the resistance to closing, which distinctly varies with line conditions; for instance, when the valve is closed with balanced pressure and there is no flow in the boiler-lead line, there is no resistance to the disk and the electrical limit must be set to trip just at the point of closing so that the disk can drift into its seat.

Assume, then, an open-end pipe condition with steam blowing to atmosphere during which the disk encounters excessive resistance. The moment the electrical limit opens, the velocity will overcome the momentum and act as a brake, thus causing the motor to stop instantly and before the valve is actually seated.

The non-return or globe valve must seat tightly and has no leeway to travel as has a gate valve. Therefore it should be made to seat tightly under all conditions and without jamming or strain.

TYPES OF GATE VALVES

The Essex station tests showed that valves of this description up to 10 in. in diameter may be safely closed under emergency conditions, and from a careful study of their behavior it seems safe to recommend them for line protection up to 12 and 14 in. in diameter. Data on emergency closing of valves of above this size and up to 22 in. are not available and the safest construction would be to machine the guides, provide the yokes with ball bearings, and attach a control unit of extra large capacity. Valves of this size are used generally for line sectionalizing, and no attempt should be made to close quickly. A typical electrically operated boiler-lead valve is shown in Fig. 5.

Table 1 shows the speeds of closing recommended under various

TABLE 1 SPEEDS RECOMMENDED FOR CLOSING HIGH-PRESSURE STEAM VALVES UNDER VARIOUS CONDITIONS

(Based on valves provided with ball bearings and machined guides. Maximum steam pressure, 350 lb. per sq. in.)

Gate Valves (Under emergency conditions, full blow to atmosphere):						
Size, inches	4	6	8	10	12	14
Type of Dean unit used	M3	M3	E1	E2	E2	E2
Closing time, seconds	20	30	30	30	45	60
Gate Valves (Under normal operating conditions, steam velocity not exceeding 15,000 ft. per min.):						
Size, inches	16	18	..	..	..	..
Type of Dean unit used	E2	E2	..	..	..	..
Closing time, minutes	2	3	..	..	..	..
Automatic Stop and Check Valves and Globe Valves (Emergency conditions):						
Size, inches	6	8	10	12[1]	14[1]	..
Type of Dean unit used	M3	E1	E2	E2	E2	..
Closing time, seconds	30	30	30	45	60	..

[1] Closes with pressure.

conditions that have been standardized by most of the valve manufacturers.

For emergency closing with an unknown direction of steam flow in the line, there seems to be little choice as to the type of gate-valve construction between the solid-wedge taper seat, the split-wedge taper seat, and the parallel-seat Hopkinson type. A valve of each type was tested at Essex station and examination showed that none of them was excessively affected by the strain.

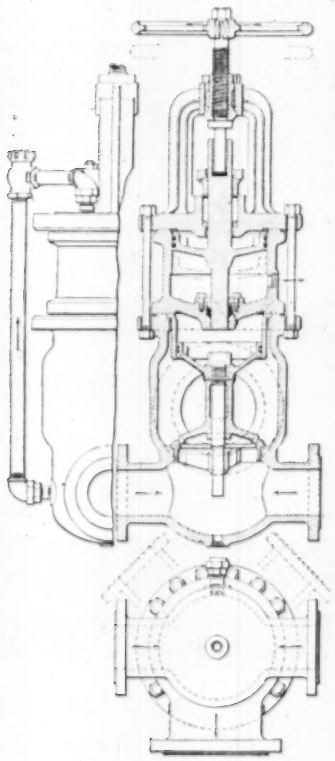

FIG. 3 CROSS-SECTION OF AN EMERGENCY PISTON-OPERATED STEAM STOP VALVE

For steam lines with one direction of flow, such as boiler and turbine leads, the Borsig type is particularly well adapted as the steam flow assists in closing and against a minimum of friction. It has the advantage of minimum pressure drop on account of the straight opening and may be quickly closed by motor operation up to large diameters. The motor-control apparatus need be only of sufficient capacity to open the valve under balanced conditions. This type is used extensively abroad in place of globe valves and a 6-in. Borsig valve that was tested at Essex operated satisfactorily under emergency conditions.

MOTOR CONTROL

There is hardly a piece of machinery in the power plant that is subjected to such extreme operating conditions as the motor-controlled valve:

a It is usually located in an out-of-the-way gallery and covered with coal dust, and in a temperature ranging between 150 and 165 deg. fahr. which necessitates special heatproof motor-coil insulation

b It is thrown directly across the line without the aid of starting resistance so that the full voltage is impressed upon the windings, and at the same moment the motor may be called upon to exert its maximum mechanical effort in unseating a tightly seated valve

c It must be small and compact, automatically lubricated, and entirely steamproof and waterproof

d Periods of inoperation range from a minimum of one week to, say, three months, during which time plant conditions may be such as to prevent its operation; and it is expected to function perfectly, even though it has not been tested over this extended period

e Steam from a leaky gland may be allowed to blow freely on the control unit for an indefinite period, and to withstand this it must be impregnated

f In case of piping rupture it is more than likely to be enveloped in condensed steam, and just at this time it will be called upon to exert its maximum mechanical effort to close the valve.

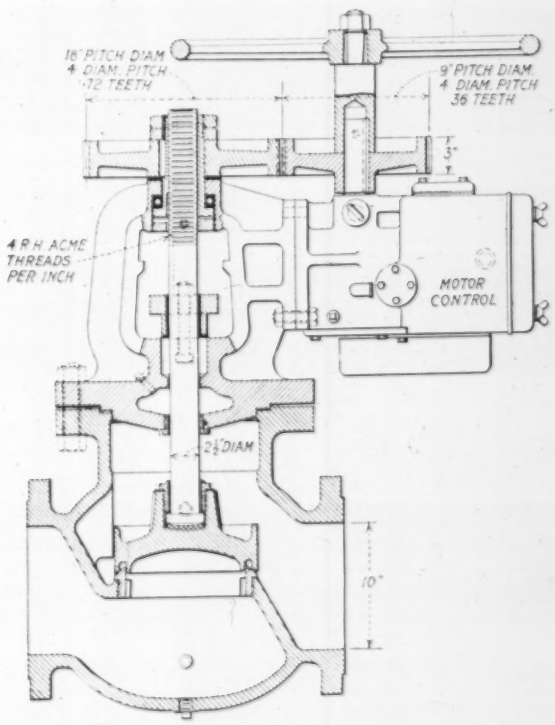

FIG. 4 CROSS-SECTION OF AN ELECTRICALLY OPERATED GLOBE VALVE PROVIDED WITH BALL THRUST BEARINGS

The duty on such equipment is extremely severe and without doubt necessitates much special construction. The motor must be of the totally enclosed type and provided with ball bearings of excessive diameter. It should be bolted directly to the reduction-gear casing, leaving no coupling or high-speed shafts and bearings exposed. It should be series wound with such characteristics that it will automatically slow down upon the application of load caused by the valve disk passing into the seat. This is the surest means of protection against cutting of the disk and seat rings.

The reduction gears should be of heat-treated steel, on account of their compactness and the excessive strain that may be applied and should be continually immersed in oil. Any oilless or self-lubricating type of bearing for either the motor or slow-speed shaft should be avoided on account of its inability to withstand excessive heat, long periods of rest, and the accumulation of dust in the bearing.

On account of the excessive drift of the motor armature and gears after tripping, a declutching device should be provided to enable the valve stem to be stopped automatically at a predetermined point without jamming. In the same casing should be provided a self-contained electrical limit which is capable of breaking the main motor current without arcing and so geared that operation of the valve by hand will not throw it out of adjustment.

The casing should be provided with one conduit opening arranged for making a steamproof joint between the internal connections and the line and all internal connections should be of asbestos-covered heatproof cable.

Control of the valves may be effected from either one point or from multiple points by the aid of a totally enclosed dustproof reversible drum switch capable of reversing the armature connections.

The stations should be provided with lamp indication showing the full open and closed positions of the valve at all times. Such a control switch should be provided with a lock, an emergency glass window, and hammer. For multipoint control an interlocking system should be provided so that the first operator to reach a valve will have full control even though other control stations are operated at the same time and in reverse directions.

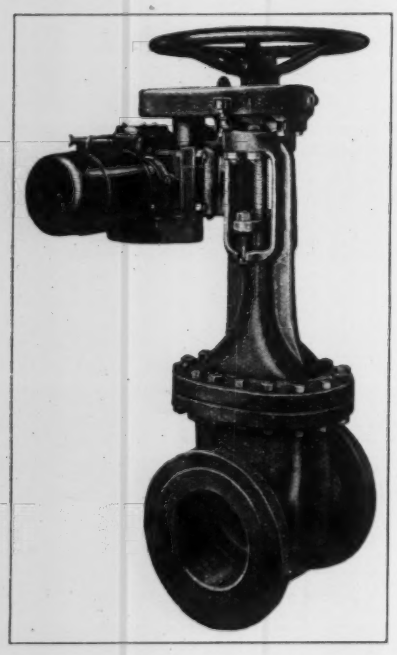

FIG. 5 TYPICAL ELECTRICALLY OPERATED BOILER-LEAD VALVE

In every case both poles of the electrical circuit should be interrupted when the switches are in the "off" position. Single-pole break is dangerous, as a ground may cause the valve to either open or close and produce very serious inconvenience.

Voltage. A valve control system should be connected to the most reliable source of current in the station, which is either the direct current exciter or switch control battery circuit, as it is equal in importance to any piece of auxiliary machinery. The voltage should not exceed 250, firstly because direct-current battery circuits of this voltage are standard, and secondly, because higher voltages would cause an unnecessary strain on the very closely packed windings. Valve motors are thrown directly across the line without resistance and encounter at the same moment their full mechanical strain due to a tightly seated valve.

Control-Station Location. Each valve should be provided with a local control station, mounted at some point from which opening and closing the valve may be observed. Preferably there should be two points of control for each valve, one in the boiler room close to the door leading into the turbine room, and the other at a safe point unlikely to be affected by steam flow. Where a centralized control board is adopted, the second point of control may be mounted in the turbine room, and for the closing of the turbine lead valves the control may be mounted on the turbine gage board.

Conduit and Cable. All conductors should be suitably protected by fuses having a capacity equal to the full-load starting current of the motor, and fuses rather than switches should be provided for disconnection. The cable should be chosen according to the location, and for the operation of high-pressure super-heated steam equipment, cable should have an asbestos heatproof covering.

It is a good plan to end the rigid conduit within 3 ft. or so of the control unit, and from that point use a flexible loop of Greenfield or similar duct. This will allow considerable vibration in the pipe line without affecting the cables, and will further enable the control unit to be freely moved.

Hand Operation. Handwheels should be provided in case of failure of current supply. No declutching device should be installed between the motor and valve which destroys the relative position of valve and motor mechanism. A properly designed and cared-for valve-operating system, especially in a power plant, need never fail, and in case of possible failure at a critical moment the valve may be closed by the handwheel, through the gears and motor, in almost as short a time as when the parts are disconnected. The great danger of a declutching device is that it is very apt to be left disconnected with no means of showing its position at the control station, and that it introduces a non-foolproof feature by providing a piece of mechanism that can be tampered with by any unauthorized person.

If such a device is insisted upon, it should be a self-contained part of the mechanism and operated externally by the aid of a wrench. Further, it must be so arranged that positive indication of its position must be noted on the indicating lights at the control station.

Periodic Tests. Every electrically operated valve should be opened and closed at least once a week, preferably under load conditions if possible, and an examination of the internal parts should be made at least once a month and all records of such tests and examinations kept on file.

THE AUTHOR'S CONTROL SYSTEM

The Dean control system of valve operation was designed exclusively for the operation of valves, and since it embodies all the

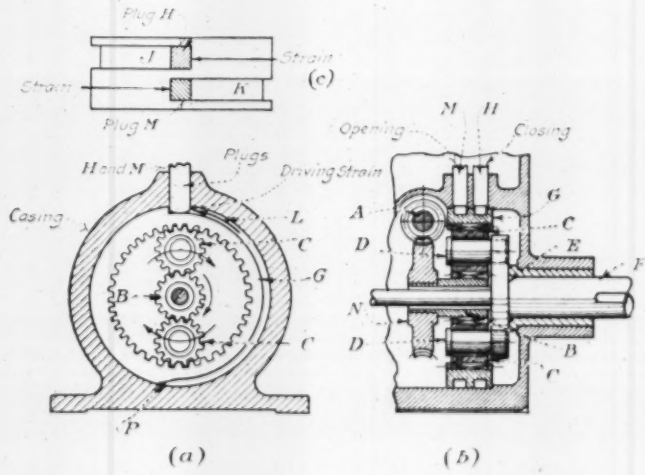

FIG. 6 MECHANISM OF ONE TYPE OF THE AUTHOR'S VALVE-CONTROL UNIT

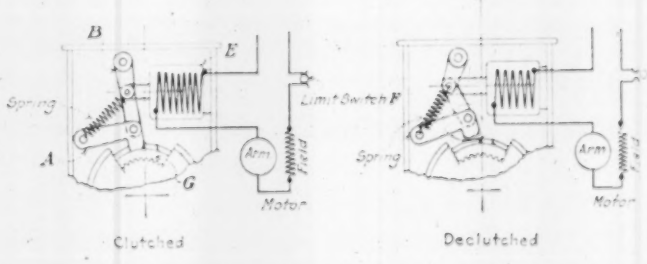

FIG. 7 SKETCH SHOWING PRINCIPLE OF OPERATION OF A SECOND TYPE OF VALVE-CONTROL UNIT

features that have been enumerated, only a brief description of its mechanical operation will be given.

The type E unit consists of a driving motor, reduction gears, and limit trip mechanism mounted in a single case. The feet of the unit are provided with four holding-bolt holes of standard dimensions for a given range of valves. Four standard units cover the complete line of valve sizes in use.

The motor is totally enclosed and steamproof, and possesses an extremely high torque. It is connected to a system of combination worm and planetary gearing for the necessary reduction in speed. The motor speed is normally 1600 r.p.m. and that of the slow-speed shaft 40 r.p.m., the gear reduction being 40 to 1.

The motor shaft is coupled to the worm shaft *A* (see Fig. 6) both running in ball bearings, and the worm wheel *N* is keyed to a sleeve forming the sun pinion *B* of a planetary gear. The planet pinions *C* revolve on large studs *D* forced into a flange *E*

forming part of the slow-speed driving shaft F, and these pinions mesh into the internal gear G forming the outer member of the planetary.

Fig. 6 (a) shows the assembly of the planetary gear, the external member or internal gear of which is marked G. The center or sun pinion B is caused to revolve by the motor, and this pinion turns the planetary pinions C and shaft F in the direction shown. The outer member or internal gear G is free to revolve on a machined surface in the casing, but is restrained from doing so by two plugs M and H which also pass through the casing in the manner shown.

Fig. 6 (c) shows another view of the outside surface of the internal gear G, and it will be seen that this surface is provided with two peripheral channels, one extending half-way around in one direction as shown at J, and the other in the opposite direction as at K. When both plugs H and M are in, or in the locked position, the internal gear G cannot move, therefore with the sun pinion being driven by the motor, the planetary pinions must turn on their own axes and travel around the internal gear, thus also moving the flange E and drive shaft F.

Referring again to Fig. 6 (a), let it be assumed that the valve is closing and the plug H is restraining the movement of the internal gear which tends to move in the direction indicated by the arrow L. At a predetermined moment when the valve is tightly closed, but not jammed, this plug is quickly withdrawn from the slot by a special cam movement, and since there is no friction now restraining the movement of the internal gear, it is of course free to revolve in the direction shown.

At the precise moment of withdrawing the plug the electrical circuit is interrupted by the limit switch and the drift of the motor coming to rest is taken by the internal gear revolving in its casing. Experiments and practice tend to show that the motor comes to rest after the internal gear has traveled about half a revolution. Should any further drift take place, the plug M is automatically moved upward by the moving tapered stop P in the groove, which operation does not open the electrical limit.

It will be noted that the plug M is still in the other slot. During the opening of the valve the motor will of course be reversed, and the internal gear will have to revolve in the opposite direction before the plug M comes in contact with the stop. This provides a hammer blow which is sometimes necessary to open a tightly seated valve, but it is by no means essential as the motor has sufficient torque with which to operate the valve under all conditions.

It is very easily seen that the moment one of the plugs is withdrawn, the slow-speed drive shaft stops instantly and that it can only stop at exactly the same point during every operation, quite irrespective of load or friction. Thus the valve gate is never jammed.

The valve gate is made to seat tightly, under power, and to a predetermined point, drift and overtravel being disregarded as a means of seating the gate.

Tachometer readings show that at the moment after tripping, the motor armature speeds up considerably before the electrical limit contact is broken, which indicates that the load is released prior to the interruption of the circuit, in this way eliminating arcing.

The gearing for speed reduction is encased in a cast-iron housing partly filled with oil, so that the gears are always well lubricated. The limit mechanism is contained in a separate part of the housing and provided with a removable cover.

The type M or magnetically operated improved Dean system as shown in Fig. 7 (a), is very similar to the type E design with the exception of an improved limit and method of releasing the restraining plugs. In place of the plugs H and M as shown in Fig. 6 (b), a restraining pawl A is provided which is held in the operating position by an energized magnet E connected in the main electrical circuit. At the moment the electrical limit switch trips at the end of travel the spring-operated toggle B is broken and the restraining pawl A instantly releases, allowing the internal gear G to freely revolve and take the complete drift of the motor armature. A very noteworthy feature in this design is that the handwheel is always free to operate without the necessity of employing a declutching device.

Discussion

N. E. Funk[1] said that the non-return valve, if motor-operated, provided the simplest scheme of boiler control. However, the practice of placing gate stop valves on the boiler header in addition to the non-return valve on the boiler was almost universal, so that it was better to operate the valve which was most readily controlled when there was any choice.

Since very definite tests had been made on gate valves showing their characteristics in so far as shutting off high-velocity steam was concerned, and since this information was not available on non-return valves, apparently the best engineering judgment would be to operate the gate valves electrically and use them to cut off the boilers in event of trouble.

There was a possibility that considerable trouble would be experienced in shutting a non-return valve against extremely high steam velocities experienced in event of the break of a header. There was considerable unbalance in this type of valve and it was necessary to shut the disk directly against the stream flow of the discharging steam. Mr. Funk believed that any attempt to use the non-return valve in this manner not only complicated the valve, with a tendency to make it less likely to function when it should act as a non-return valve, but also would mean redesigning practically all of them so that they would withstand the tremendous forces encountered when shutting them against a break.

Experience in closing a standard 250-lb. non-return valve against a 400-lb. hydrostatic test on the boiler had shown how difficult it was to make such a valve tight, it being necessary for two men with a long wrench to pull up on the hand wheel to put a heavy strain on the disk to keep the edges from lifting under this pressure and causing the valve to leak.

Trouble of course might be experienced in shutting gate valves by having the wedge plow up the seat and turn up enough metal ahead of it to prevent the gate from closing. The tests made at the Essex plant, however, seemed to indicate that not much fear of this need be experienced.

Until very recently large steam valves had been operated electrically more as a matter of convenience and labor saving than with the idea of shutting off the steam in event of a steam-line break. The old types of controls, however, would not close a valve against any unusually high steam velocity due to insufficient power in their operating mechanisms.

As to steam-pipe sectionalization, Mr. Funk stated that he had heard some very good engineers speak more or less disparagingly of such procedure, indicating that they believed that piping should be so designed that it would be impossible for a break to occur. But even if everything was tested considerably above its working pressure, there was a possibility of rupture from outside causes or from flaws which could not be detected in this manner. In addition, with the industry thinking of higher pressures and temperatures at which there was uncertainity as to what would happen to the metal, the necessity of sectionalizing was even greater.

His company believed thoroughly in remote control of steam valves, and in their new designs were not only arranging to shut off each boiler from a remote point, but also to sectionalize the steam-pipe headers. The remote-control point was located outside of the turbine and boiler rooms so that no matter where a break occured the damaged section might be isolated without a serious or lengthy interruption of operation.

W. S. Morrison[2] said that the New York Edison Co., operating at only 200 lb. pressure, had never experienced any difficulty in valve closing, but to forestall any such troubles as those described in the paper they had provided controls on their 18-in. header valves. When they came to consider the question of higher pressures, however, the subject would be analyzed a great deal closer.

Alfred Iddles[3] said that it seemed to be the general practice to consider the non-return valve purely as a check valve, and that in his opinion it should be confined to this use for the present. Later it might be possible to develop a device that would function successfully as a combined shut-off and non-return valve.

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[1] Operating Engineer, Philadelphia Electric Co., Philadelphia, Pa. Mem. A.S.M.E.

[2] Asst. Mech. Engr. N. Y. Edison Co., New York, N. Y. Mem. A.S.M.E.

[3] Mech. Engr., Day and Zimmerman, Philadelphia, Pa. Mem. A.S.M.E.

SURVEY OF ENGINEERING PROGRESS

A Review of Attainment in Mechanical Engineering and Related Fields

The Gas Turbine in Theory and Practice

A SERIES of articles discussing the subject broadly. The author considers the problem of gas-turbine development a difficult one but not impossible of accomplishment. It is primarily, in his opinion, a matter of engineering design and selection or development of suitable engineering materials. As regards the field of the gas turbine, he believes that while it will not displace in many cases the present internal-combustion motor, it will, under certain conditions, prove a formidable competitor of the steam turbine. It has, however, a long road to travel before this will really become of serious commercial importance.

As regards the competition between the gas or heavy-oil engine and the gas turbine, the former though economical when it does not exceed a certain power output, becomes unwieldy in very large sizes. The real scope for the internal-combustion turbine will be found in certain cases where, in competition with steam plants, the elimination of the boiler and condensing installations offers peculiar advantages, while in other, such as the utilization of blast-furnace gases, the fact that it can be made to develop large powers on comparatively small dimensions may, even in spite of possibly inferior thermal efficiency, render it a suitable alternative to the large gas engine. Until, however, some well-thought-out design is actually constructed and tested, engineers must rest content with estimates of efficiency instead of proofs, and it remains to be seen whether they are of a sufficiently attractive nature to justify embarkation on the development of a new type of prime mover. Without experimental verification, much further progress in the design of the gas turbine cannot possibly be made.

The main difficulty encountered by designers of gas turbines lies in the usual severe temperature conditions. In the constant-pressure internal-combustion turbine, the application of heat at its maximum is constant; both in it and in the constant-volume type the blading is subject to a constant impinging temperature, although one of reduced intensity, but in both cases the function of the material is to withstand the heat and, as far as possible, not to transmit any of it. It will immediately be realized, therefore, what considerable differences and difficulties are involved in the use of internal-combustion products, whether expanding in a cylinder against a piston or impinging upon a turbine wheel through a nozzle, for the maximum reduction in initial temperature that can efficiently be used in the latter case is only in the region of 1000 deg. fahr., and therefore the temperatures in furnace, combustion chamber, or explosion chamber, and that on the blading can only be separated by about this amount. This very high initial temperature of the working fluid at first involved very considerable mechanical difficulties, even in the intermittent working of the reciprocating oil engine. It has hitherto almost entirely frustrated the solution of the gas-turbine problem, and nearly all, if not all, the serious difficulties encountered arise from this element—the initial temperature. The constructional troubles likely to be created from this cause react on the entire proportions of the design, and in order to obviate them recourse must eventually be had to improving the heat-resisting properties of metal. At present it is necessary either to restrict the initial temperature to such a point that, with the nozzle drop, the impinging temperature—probably the cardinal point in all gas-turbine design— does not cause deterioration of the blading, or else to accept some form of intermittent working, either of which will reduce the amount of positive work obtainable from the cycle or involve a more elaborate machine. The nature of the internal-combustion cycle—to be referred to later at length—completes the chain of difficulties, because, if the turbine operates on the constant-pressure cycle, the proportion of negative work to the available positive then becomes unduly large; while if it operates on the constant-volume cycle, the positive work obtainable can only be developed in a multi-chamber machine which, compared with other forms of prime mover, is unduly large and complicated. Moreover, the real point at issue must always be borne in mind; there is no question at all of evolving an internal-combustion turbine which shall be saliently superior to the present oil engine, just as the steam turbine proved to be to the reciprocating engine; economically this is probably impossible with a non-condensable gas. The real question is whether, in spite of these limitations, and the difficulties which are liable to arise if they are exceeded to any extent, a gas turbine can be produced which will compete with the steam engine and its accessory plant.

Of all the fragmentary work which has already been done on the gas turbine very few designs and even fewer test results have actually been made public. It would appear, however, that efficiencies equal to or even exceeding the best oil-engine practice are probably obtainable in theory, though not within the same cycle limits. They are not in sight today and are unlikely to be for some years, mainly owing to the lack of material suitable for withstanding the very high temperatures which such efficiencies demand.

The original article gives construction data and results of trials of a number of internal-combustion turbines. This part is not suitable for abstracting.

Of the many various types of gas turbine which have been constructed, the majority were designed to work on the constant-pressure cycle; but of those for which any detailed information has been published, three out of the seven were explosion turbines working on a constant-volume cycle, one was an exhaust turbine, and the remaining three, while operating on the constant-pressure superatmospheric cycle, were all mixed-fluid turbines. That is, besides the air and the fuel, steam or water injection was used to reduce the temperature of the mixture. Two of these constant-pressure, mixed-fluid turbines were also tested with air only. No published results are available beyond these air tests for a pure constant-pressure single-fluid machine, nor for any gas turbine working on a subatmospheric cycle, though the latter would probably give results very much superior to any other type. Before going any further into the question, it is necessary to explore the possible efficiencies obtainable when operating the turbine form of engine on an internal-combustion cycle, and the constant-pressure method will be investigated first.

In all heat engines the continuous output of work is effected by some from of mechanism which utilizes the pressure difference in a working substance created by the heat energy supplied. Whichever cycle is adopted, however, the internal-combustion turbine, as is the case in all practical heat engines, consists of the following essential parts:

1 A furnace or combustion chamber in which heat is supplied to the working fluid.

2 A heat sump or cooler into which the working fluid expands after leaving the combustion chamber.

3 Between these, a mechanism which will absorb the available energy of expansion in the fluid as it flows from 1 to 2 and reproduce it in the form of positive work, and

4 A mechanism which may either (*a*) supply the working fluid to the combustion chamber, (*b*) withdraw it from the cooler, or (*c*) combine both functions.

The work for these may be supplied either by the excess positive work or by other means, but in any case forms the negative work of the cycle. Owing to the fact that for the heat flow from furnace to cooler—which in the case of internal-combustion engines using

non-condensable gases must eventually be the atmosphere—and because this rejection usually takes place at a pressure somewhat greater than that of the atmosphere, in the constant-pressure cycle air must be supplied to the furnace under pressure, whereas the constant-volume cycle can, if necessary, be worked without precompression, although a certain small amount is advisable in practice. In lieu, however, of using a purely superatmospheric cycle, either an entirely subatmospheric cycle can be adopted, an exhauster taking the place of the compressor, or a combination of the two may be adopted. Precompression has hitherto always been adopted, the subatmospheric cycle not having been employed, and with the exception of Holzwarth, who uses a very small degree of exhaustion, the exhaust pressure has hitherto been above atmosphere. It was stated above that the constructional difficulties all arose through the element of temperature. The very real difficulty in achieving an efficient design of internal-combustion turbine, which would render it *facile princeps* among prime movers, lies in this matter of compression or exhaustion—that is, in the negative work of the cycle.

It should be clearly emphasized at this stage of the thesis that the broad principles upon which heat energy can be supplied, absorbed, and rejected in their various proportions are perfectly well known, and that no design which fails to comply with them can possible succeed. This is the reason why a good working solution of the internal-combustion-turbine problem depends in no way on some fortuitous or radically novel design on the part of some lucky inventor.

Owing to the initial temperature difficulty, the reaction turbine is unsuitable for the internal-combustion cycle, though it may become feasible eventually for the exhaust and in the case of certain forms of subatmospheric working, but only if very large volumes are involved. Our choice is therefore restricted to either the single-wheel impulse turbine or to a velocity-compounded turbine with two, or possibly three, rows of moving blades, though it is very doubtful if the efficiency of the latter is even nearly adequate. From the cycle limits and the analysis of the working fluid, it is possible to calculate the maximum value of the available energy, and the turbine efficiency is the ratio between the useful work developed in the turbine shaft and the energy available for expansion. Two transformations of energy take place in the turbine: first, from thermal to kinetic energy; secondly, from kinetic energy to useful work. The first loss which occurs, lies in the radiation from the combustion chamber, which reduces the amount of energy available for conversion into kinetic work; the second consists in the friction and eddy losses in the nozzle. From this point the author proceeds to a discussion of the various losses.

As regards the stage efficiency of the turbine itself, the author bases his opinion on curves of stage efficiency for various ratios of blade speed to steam speed for different angles of wheel for which data have been published by Lasche and Baumann. From this it would appear that within very narrow limits the respective stage efficiencies do not exceed 0.8 and 0.68. All things considered, the author comes to the conclusion that for gas-turbine work the two-row wheel is practically the only one suitable.

It would therefore appear that the successful operation of a gas-turbine, apart from questions of mechanical construction, depends primarily on: (1) The ratio of positive to negative work in the cycle employed, and (2) the efficiency of compression and expansion, and for the constant-pressure superatmospheric-cycle single-fluid machine it was shown that with a two-row velocity-compounded turbine of 65 per cent efficiency, the ideal brake efficiency, even with an initial pressure of 500 lb. per sq. in. pressure, would not exceed 19.6 if the compression were adiabatic and the compressor had an efficiency of 90 per cent, or 23.2 if the compression were isothermal and the efficiency reached 70 per cent. These brake efficiencies are capable of considerable improvement on the air cycle used, but when dealing with the real working fluid the absolute thermal efficiency is adversely affected by the variation in the specific heat of the substance.

Compared with the internal-combustion piston engine, the gas turbine can never enjoy the vast advantages over the reciprocating type of mechanism which the steam turbine possesses. The same brake thermal efficiency may be obtained by working comparatively inefficiently on a cycle of high absolute efficiency just as well as by working more efficiently on a cycle of lower absolute efficiency. This is precisely what occurs in the case of the steam turbine and the piston engine. With steam, however, enormous expansion ratios are readily obtainable, in fact, a volumetric expansion of 750 to 1 is not out of the question, and the turbine mechanism lends itself very readily indeed to the utilization of small pressure differences and the "toe" of the *P-V* diagram. It can also utilize much higher superheat than can the engine, for there are no surfaces in contact, as well as vacuums which are entirely beyond the range of constructional possibilities where piston and valve mechanism are involved. Increased thermal and mechanical losses with reciprocating mechanism follow any attempt to go far in the direction of utilizing the energy of expansion at pressures below about 2 lb. abs., though there is still useful energy remaining in the steam. The result is that this engine is restricted to a cycle

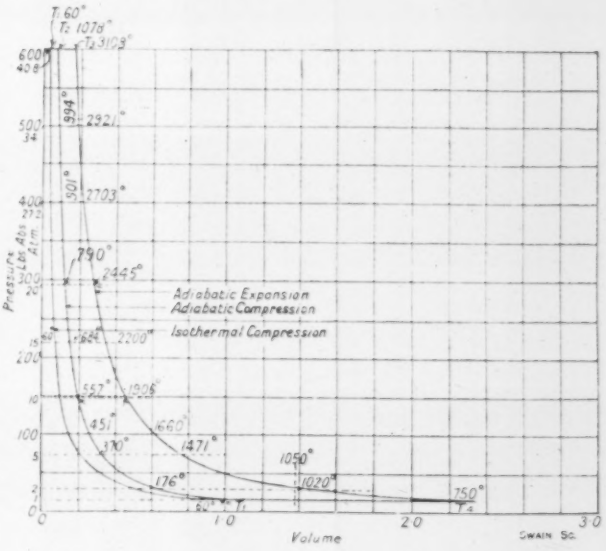

FIG. 1 PRESSURE-VOLUME DIAGRAMS FOR VARIOUS CYCLES

of much lower efficiency In the internal-combustion piston engine, as shown in Fig. 1, the "toe" of the diagram with atmospheric exhaust only forms a small proportion of the available work, this fraction becoming smaller as the initial pressure increases. But with the products of internal combustion, which are non-condensable gases, the advantages of the turbine over the piston engine, in that it can utilize the "toe" of the diagram, very largely vanish, because the expansion ratio is greatly curtailed by the fact that the gas does not change state, and though use can be made of subatmospheric working, the limits of the internal-combustion *P-V* diagram cannot be extended in the same manner as they are with steam owing to the condensation which the latter undergoes. We are, in the internal-combustion turbine, as with steam, employing a mechanism of lower intrinsic efficiency with which to extract the available kinetic energy, but whereas in the latter the limits between which we can readily work are so much greater, and consequently so much more energy per pound of working fluid is available, in the former the range of expansion available hardly differs at all in the two cases, and pressure compression ratios in both exceeding 15 offer hardly any advantages with the materials at present at our disposal, as higher compressions involve initial temperatures which become increasingly difficult to handle. The efficiency of a non-condensing steam turbine is notoriously low, and between initial and atmospheric exhaust pressures the piston engine is much superior—is even superior down to, perhaps, 26 in. vacuum. The same applies with internal-combustion gases, and no little misapprehension has often been caused by lack of appreciation of the fact that the turbine per se is inherently less efficient than the reciprocating engine except in the utilization of the subatmospheric range, but that it derives its vastly superior brake thermal efficiency from the fact that it can operate on a cycle of far higher absolute thermal efficiency, and *not* from the fact that its mechanical efficiency or efficiency of energy conversion is better, because it is not so except at very low pressures.

With the limit of velocity-compounded-turbine efficiency on the

low side of 70 per cent there are two directions in which improvement in the efficiency in internal-combustion-turbine installations can be made, viz.: (1) By increasing the air-compressor efficiency, and (2) by an increase in the ratio of positive to negative work in the cycle. The former improves the thermodynamic efficiency of the combination, and therefore the brake thermal efficiency for a given cycle. *The Engineer*, vol. 135, nos. 3514, 3515, 3516, 3517, May 4, 11, 18 and 25, 1923, pp. 466–468, 490–491, 515–517, and 557–559, 7 fig., 12 tables.

Short Abstracts of the Month

AIRCRAFT ENGINEERING

A New British Light-Weight Airplane

The Gnosspelius Light Plane. General description of the airplane designed by Maj. O. T. Gnosspelius and recently tested at the Lympne aerodrome.

The design of the plane is based on aerodynamic tests made with the pendulum apparatus which was developed at the works of Short Brothers. On this pendulum the models are mounted with the wings parallel to the pendulum arm, and the tests are carried out by raising the pendulum to a horizontal position, releasing it from there and noting how far it swings up on the other side of the dead center. This gives an indication of the resistance of the model, while the lateral deflection of the pendulum at its lowest position gives an indication of the lift.

On this pendulum Major Gnosspelius has tested a large number of models, and he has discovered several very interesting, and to some extent inexplicable, things. First of all he tested certain wing sections of which data were already available. He found that reasonably good agreement was obtained. He then tested other wing sections, as, for instance, the R.A.F. 19, which was found by model tests at the National Physical Laboratory to give a very high lift. On the pendulum this high lift was not attained. Now the curious fact about this test is that when tested on the full scale R.A.F. 19 does not give the high lift which wind-tunnel tests on models indicated. The logical conclusion seems to be that the pendulum, in certain circumstances, is in better agreement with full-scale work than is the wind tunnel. A good deal more evidence is necessary before one can definitely say that this is so, but several experiments seem to point that way. If this should prove to be so, the fact may considerably alter present views on aerodynamics, and it even seems possible that the phenomenon commonly known as "scale effect" does not exist, and that the differences between model and full-scale results, as indicated by wind-tunnel tests, may be due to other causes, possibly to the effect of the walls of the wind tunnel.

The Gnosspelius light plane has considerable resemblance to a bird in so far as the shape of the wings, body and tail are concerned. The body, which is of circular cross-section, comes to a point in the nose, and this point is not on the center line but is slightly bent down much as is the head of a bird during flight. The tail plane is long in relation to its span and is of triangular plan form. The ailerons are of considerable size, each being over 12 ft. long and of high aspect ratio.

The wing section employed is of fairly deep camber, although not of great thickness. One result of this is that it would have been almost impossible to obtain sufficient torsional stiffness with the usual two-spar construction. Consequently Major Gnosspelius decided to employ four spars, and so to design his ribs that any deflection of any one spar should be transmitted to the other three. The spars themselves are of the box type, with spruce flanges and three-ply walls. The ribs are of lattice construction, with double flange strips, in between which are passed the ends of the lattice bars, tacked and glued in place. To be quite accurate, it should, perhaps, be said that the ribs are N-girders rather than lattices, as the compression members are vertical.

The plane weighs about 530 lb., which figure includes gasoline or about two hours' flight, and the minimum horsepower required or horizontal flight is in the neighborhood of 3 b.hp.

The machine is fitted with specially light glider instruments such as an air-speed indicator reading from 10 to 80 m.p.h., a revolution indicator, an aneroid, and oil-pressure gages.

In tests during the first flight a height of about 1500 ft. was reached, the machine climbing strongly. The duration of this flight was about 15 min. In the next flight a height of 2500 ft. was reached and the machine remained in the air for approximately half an hour. The maximum horizontal speed attained as shown by the air-speed indicator was about 65 m.p.h. (*Flight*, vol. 15, no. 753/22, May 31, 1923, pp. 292–295 and 291, 8 figs., *d*)

AIR MACHINERY

Explosions in Air Compressors

Explosions in Air Compressors, J. A. Vaughan. Discussion based on official inquiries into sixteen explosions occurring in the pipes of passages through which the gas passed. The author comes to the following conclusions:

a Carbonaceous deposit is the seat of the trouble.

b Leaks in the final stage of compression or discharge are dangerous.

c The most likely time for firing or explosion is when the compressor is unloading.

d One or two fusible plugs do not provide an adequate safeguard against firing, explosion, or the delivery of dangerous air to the mine workings.

e The system of temperature gaging and record requires alteration.

The term "carbonaceous deposit" is used to denote the deposit which is left in the intercooler system, the valve chests, or the discharge passages owing to the cooling and the baffling influences to which the moving compressed air is subjected.

The proximate analyses of the deposits show that the substance, besides being a good gas reservoir, is of a nature liable to spontaneous combustion, while it is probable that the presence of metals in a finely divided state accelerates the oxidizing process, these metals being maintained in an unoxidized condition owing to a protecting film of oil.

There is little doubt that, as is the case with coal, the rate of absorption of oxygen rises with the temperature, so that, seeing that the temperature of the carbonaceous mass is necessarily increased as the oxidation proceeds, the rate of absorption will increase by a sort of compound-interest law, in the absence of any cooling effects.

The discharge pipes and receivers are unlagged and free to radiate heat, while at full load the stream of compressed air (at, say, 300 deg. fahr.) must assist in retarding any tendency of the deposit to heat up to a dangerous degree, unless the deposit is situated in mass, out of the flow, as may be the case in the bottom ends of intercoolers.

As regards the leaky discharge as a source of danger, the author shows by a series of indicator cards the effect of a leak, which, in this case, was intentionally created in the valves or piston of either the first or second stage of a 1440-cu-ft. air compressor. The inspection of the diagrams shows material alterations from normal conditions. Among other things, the fact that in a two-stage compressor leakage in the high-pressure discharge valves or in the piston will always raise the first-stage discharge and intercooler pressures. The temperature of the compressed air in this position will always necessarily be higher.

At full load the effect of the high-pressure discharge-valve leak is to put more work on to the low-pressure side. In this cylinder, with its inlet a little delayed due to the clearance air being of a higher pressure, a full charge of air at atmospheric pressure is compressed to a pressure higher than normal, the actual height reached depending on the size of the leak, which determines the point of the high-pressure stroke at which the inlet valve will open. The capacity of an intercooler of usual proportions will not be sufficient to prevent a rise of temperature on this account. If the leak were sufficiently large the high-pressure piston would merely float in air at receiver pressure, while the whole of the compression would be accomplished in the first-stage cylinder.

In addition, the author presents the curve giving data of tests with a broken valve plate in the second-stage discharge, from which it would appear that:

a Heating undoubtedly does take place if a high-pressure discharge valve leaks.

b The smaller the load, the greater the heating effect under this system of governing.

c The higher the pressure, the greater the heating effect.

There are two causes, interlinked, why leaky discharge valves produce rise of temperature in the receiver air (meaning by this term the compressed air that is in the discharge pipe or receiver—outside the high-pressure cylinder):

a The rate of compression in the low-pressure cylinder is increased, as more work is performed by this cylinder and a higher delivery pressure reached. Therefore the high-pressure intake is at a higher pressure and temperature, and consequently the high-pressure delivery and the receiver air are higher in temperature.

b The air that leaks back into the high-pressure cylinder does not fall appreciably in temperature on reexpanding, and is by reason of its recompression delivered back to the receiver at an increased temperature.

As regards the danger at the time of unloading, the author advances the point that this time appears to be the most suitable for the proper explosive mixtures to be formed. Further, the dangerous gases being formed at the compressor end of the line should be capable of detection. An alarm device which would call the attention of the engineer to danger in that carbon monoxide or methane were being generated, should prove effective in preventing gassing or explosion. (*The Journal of the South African Institution of Engineers*, vol. 21, no. 9, April, 1923, pp. 171–176, 7 figs., *pA*)

ENGINEERING MATERIALS

TENTATIVE SPECIFICATION FOR GRAY-IRON CASTINGS. Text of the specifications prepared for submission to the Manchester Conference of the Institution of British Foundrymen in June, 1923.

All gray castings are divided into two classes: important castings, in which test bars shall be cast on when possible; and other castings, in which the bar may be cast separately at the option of the foundryman.

The thicknesses of any casting shall determine the size of the test bar required, namely:

Bar 0.875 in. diam. cast 15 in. long, for all castings where no main cross-section of the metal exceeds $^3/_4$ in. in thickness.

Bar 1.2 in. diam. cast 21 in. long, for all castings where no main cross-section of the metal exceeds 2 in. in thickness.

Bar 2.2 in. diam. cast 21 in. long, for all castings where no main cross-section of the metal is less than 2 in. in thickness.

The specification gives requirements for the transverse test and the tensile test, and provides that where physical tests are specified the amount of any chemical element other than phosphorus shall not be specified, and for phosphorus only its maximum percentage. (*The Foundry Trade Journal*, vol. 27, no. 354, May 31, 1923, p. 449, *p*)

PEARLITIC CAST IRON. Pearlitic cast iron is a cast iron having a microstructure consisting mainly of pearlite with deposited graphite. The properties claimed for it are said to be:

1 High transverse and tensile strengths and toughness

2 High resistance to impact stresses

3 Moderate hardness when properly treated

4 Only a slight tendency to the formation of "pipes," and hence the possibility of making complicated castings

5 Great resistance to sliding friction (abrasion)

6 Fine and dense structure, the structure being unaffected by temperature changes.

The process of producing pearlitic cast iron consists essentially in the combination of two media, in the variation of the furnace charge, and in the correct heat treatment of the molds. Correct mixing is intended to give as little stimulus as possible to the formation of graphite. The materials used are carbon, silicon, and low-phosphorus pig iron. High sulphur content does not appear to be detrimental.

When cooled in the ordinary way, an iron of this kind would solidify "white." In order to preserve the desired pearlite-graphite structure a slow rate of cooling is necessary, this being obtained by preheating the mold. The degree of preheating depends of course on the wall thickness of each particular casting.

Theoretically speaking, it would be possible to get, from one and the same charge, and by suitable heat treatment of the mold, every desired cross-sectional area with the same final product (pearlite-graphite structure) when once the rates of cooling for the different cross-sectional areas had been determined. In practice, various cross-sectional areas are grouped together and a special burden or charge selected for each group, assuming the same amount of preheating of the mold. The possibilities of pearlitic cast iron have been discussed by Professor Bauer in a recent issue of *Stahl und Eisen*. The present writer has carried out some tests with the view to determining whether the claims for this new type of cast iron are justified and gives the results in the original article. From these it would appear that pearlitic cast iron has superior strength properties as compared with other kinds of cast iron tested. It would further appear that the casting made by the pearlitic cast-iron process must be practically free from stresses owing to its slower and more uniform cooling, whereas the ordinary types of cast iron show considerable strains due to casting. (*The Foundry Trade Journal*, vol. 27, no 355, June 7, 1923, pp. 454–456, 5 figs., *dp*)

FUELS AND FIRING

The Clyde System of Oil-Fuel Burning

THE CLYDE OIL-FUEL BURNING SYSTEM. Description of the equipment fitted on the *Fezara*, a geared-turbine steamship of the Khedivial Mail Line.

The outstanding features of the Clyde oil-fuel burning system are a compact and novel design of heater and pump unit, a simple

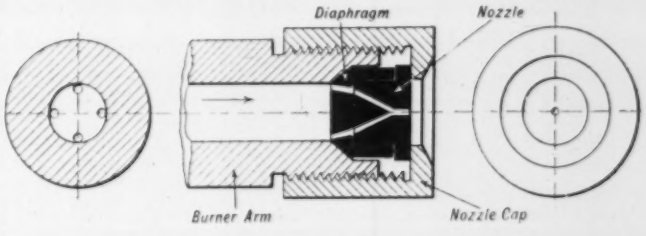

FIG. 1 ARRANGEMENT OF ATOMIZER

oil-spraying device, and a special arrangement of furnace front, which, with suitable modification, may be fitted to any type of boiler.

In this system the fuel is drawn from the storage or settling tanks and suction strainer by a pump, which delivers it to a heater in which its temperature and corresponding viscosity are raised to the point necessary for its complete atomization under pressure in the burner, through which it is injected into the furnace in the form of a finely atomized conical spray.

As regards the heater, the flow of the oil being in series through the tubes, the danger of silting up is obviated, for the scouring action set up sweeps the tube spaces clear and prevents any deposit taking place. But, should one tube give out, only a small part of the whole heating surface is rendered inoperative. The main heater body is made of cast iron, and has within it a series of bored holes which are interconnected at their ends by means of machined passages. The arrangement of these passages compels the oil to travel in a sinuous path throughout the entire length of the heater.

The atomizer used with the Clyde pressure-jet system is shown by Fig. 1. The patented spraying nozzle is carried on the lower extremity of an L-shaped arm which is clamped by a quick-release attachment to a fixed bracket on the furnace door. Both the diaphragms and nozzles have been designed for high efficiency. They impart to the oil projected into the furnace the centrifugal force necessary to break it up into a minutely atomized conical spray, in which condition it is in a position to combine intimately with the air required for combustion. The atomizing effect is produced by a series of four oblique holes converging toward a conical projection, and a feature of the design of this part is that the conical projection is completely shrouded, thus protecting the point from injury.

The spraying nozzles are so proportioned as to be capable of

completely atomizing the oil at pressures of 25 lb. per sq. in. and upward. The weight of the oil dealt with and the output of the burner may be varied through a wide range by increasing or decreasing the pressure. When working in conjunction with the air director, the contour of the flame produced may be kept constant throughout the pressure range for the particular size of nozzle which is being used. Should it become necessary to work at pressures either above or below the normal limit, it is only necessary to substitute another size of nozzle corresponding to the requirements of the new pressure. Only the nozzle need be replaced, and the diaphragm remains the same throughout all pressure ranges.

The furnace fronts are described in the original article. As regards the starting-up gear, which is an important element when no steam is available, a special steam boiler fired by a Clyde kerosene-vapor burner is employed, the steam generated in this boiler being used to drive one of the pumping and heating plants.

The original article gives data of the results obtained with the oil-fuel system described and also the steam and air systems which may be used in place of the pressure-jet system.

The Clyde steam-jet system of oil burning is not quite as economical as the pressure-jet system, but is cheaper in first cost. The low-pressure air system is especially applicable to small boilers, galley ranges, central heating, and general furnace work, the oil being atomized by air at a pressure varying from 8 in. to 18 in. water gage. The Clyde high-pressure air burners may be used in cases where intense local heat is required. None of these systems is described in any detail. (*The Engineer*, vol. 135, no. 3519, June 8, 1923, pp. 611–614, 15 figs., *d*)

INTERNAL-COMBUSTION ENGINEERING (See Measuring Apparatus)

MACHINE PARTS AND DESIGN

Avoiding Shock on Intermittent Machine Work

Avoiding Shock on Intermittent Machine Work, Fred Horner. Discussion of methods of overcoming the jerky action when tools are cutting ridges or passing over slots and gaps, and also for preventing backlash in gears.

After describing some of the cruder methods the author mentions types of electric drive which provide control by tappets appropriately situated at the table edge by which the motor speed is increased while the tool is traveling over the empty spaces, and brought to normal cutting speed just before encountering the metal again.

The same principle is now being applied to the operation of some milling machines, feed increases being arranged to jump the work gaps at high speed.

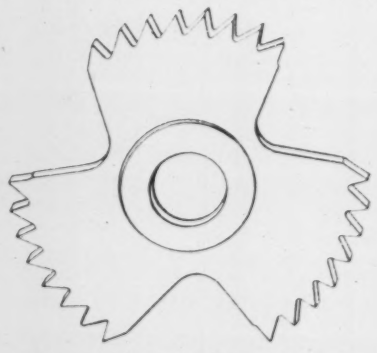

Fig. 2 Relieved Slitting Saw

The general object of such tools is to provide a relief from the strenuous continual action of normal tools, either in the way of dividing the chips into smaller portions, or of that idea combined with greater facility for chip escape. The principle embodied means omission of a succession of similarly shaped cutting edges, and their staggering or modification of lateral position, or the alteration of alternate teeth in such a way that successive chips differ.

The commonest model on the latter plan is the well-known stepped cutter used for gear cutting, which has nicked teeth with staggered grooves, and the cutter with alternate plain and waved teeth. A good many milling or slitting saws and the larger saws used on the regular cold-sawing machines are built with teeth having interrupted action, such as every other tooth a vee, splitting the square or round-nosed cut of the normal teeth. Or alternate teeth are beveled opposite ways, to throw the chips out freely. A somewhat similar incident occurs with the alternate-tooth mills now utilized for cutting deep grooves, the teeth standing to the right and left so that ample space for cutting and coolant is left in front of each tooth.

A novel principle has also been applied to slitting saws (Fig. 2) giving relief from the continuous cut all around and letting coolant get in and chips come out with greater ease. This scavenging action of the open spaces enables deep slots to be rapidly cut without fear of saw breakage. (*Canadian Machinery*, vol. 59, no. 22, May 31, 1923, pp. 17 and 24, 2 figs., *dt*)

Nomenclature of Power-Press Parts. The original article gives two drawings of a power press in which each part is numbered, together with a list of the names for the various parts. The availability of such a list is desirable as various terms are used to identify the same style of press; as, for example, "straight column," "straight-sided," and "straight-sided pillar." Even the machine itself is sometimes called a power press and sometimes a punch press; in the latter case it may be confused with the machine for punching holes in plates.

The nomenclature reproduced in the article is taken from a catalog issued by the Toledo Machine & Tool Co., Toledo, Ohio. (*Machinery* (N. Y.), vol. 29, no. 10, June, 1923, pp. 813, *p*)

MACHINE TOOLS

A Novel Combination Bench Tool

The Walker Mechanic. Description of a tool which aims at combining the features of a lathe, a drilling machine, and a vise.

The base of the machine comprises a casting which has more

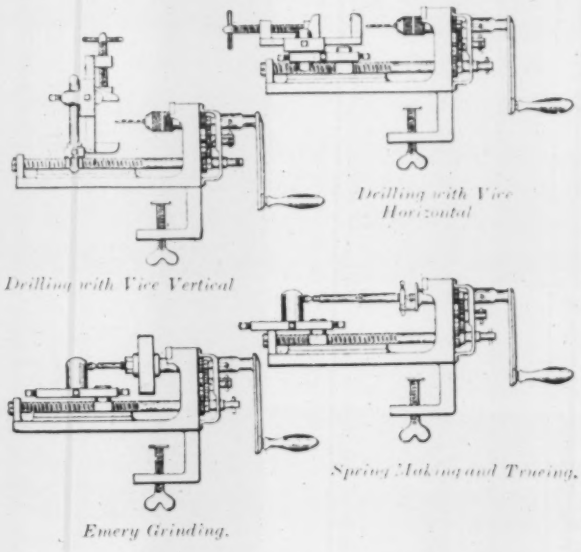

Fig. 3 The Walker Mechanic

or less the form of the front jaw of a parallel-jawed vise. This part, which is clamped on to the bench, is equipped with two short spindles and a long screw that are all geared together. The driving-crank handle can be fixed on any of the three spindles, so that a corresponding number of speeds can be obtained.

The illustrations show how, by means of some comparatively simple adapters, the machine can be employed in several different directions, while a substantial back jaw that is provided can be used in combination with the base and screw to form a powerful vise. The back jaw, incidentally, has a fair-sized anvil face and beak. (*The Practical Engineer*, vol. 67, no. 1894, June 14, 1923, p. 335, 1 fig., *d*)

MARINE ENGINEERING (See Fuels and Firing)

MEASURING APPARATUS

Positive Measurement of Fuel Oil on Motorships. Description of the Bassler meter, which integrates the flow through oil lines with compensation for temperature.

The meter may be calibrated either in "constant-weight gallons"

or in "constant-volume gallons" and in both cases it is, within practical bounds, unaffected by rates of flow, line pressures, viscosities, temperatures, or other variable factors.

In construction this device consists fundamentally of a closed cylinder containing the oil, within which there is a concentric barrel, through which the oil is measured by the ton. The space between the measuring barrel and the outer body is divided into two chambers, one containing the inlet valves to the measuring barrel and the other housing the delivery valves. Thus the flow through the meter is always in one direction.

Piston valves are used to which plunger rods are attached. At a certain point in each stroke the measuring piston makes contact with the plunger rods of the inlet valve and delivery valve at one end of the cylinder and moves them against spring pressure until, at a predetermined point, a trip falls and holds the valves in place. Simultaneously, on the other side of the piston, a trip is released, permitting the pair of valves at the opposite end of the cylinder to assume their defined position under their spring action. It will be readily understood that the valve operation requires the inlet valve on one side of the piston to be open when the inlet valve on the other side is closed, and the delivery valves to be correspondingly taken care of. There is nothing special or intricate in this.

Of importance, and of particular design, are the adjustments, two in number, namely, the thermostatic control and the manual control. The first of these consists of a small metal tube containing a liquid of similar coefficient of expansion as the liquid that is to be metered. It is permanently fixed at one end, with the other end floating and carrying the valve-trip bar. Consequently the temperature changes cause the valves at one end of the meter to open proportionately earlier or later, thus having the effect of increasing or reducing the volume measured by the piston on each stroke. In this way the measurement by weight-gallon is maintained constant. Manual control is virtually a factory adjustment, being made on the valve trip at the other end of the measuring cylinder.

For any definite grade of oil the thermostatic control can be made exact. To take care of the varying grades of oil which a ship may have to purchase as it bunkers at different parts, the engineer should order a series of thermostatic controls to be charged with different liquid at the factory. (*Motorship*, vol. 8, no. 6, June, 1923, pp. 426–427, *d*)

MOTOR-CAR ENGINEERING (See Special Processes)

EIGHT-IN-LINE ENGINE PACKARD CHASSIS, J. Edward Schipper. The new Packard chassis incorporates two features novel in American motor-car construction, namely, the eight-in-line engine and four-wheel brakes. The eight-in-line engine has been selected to replace the twin six because of its overlapping power impulses, its inherently good balance with consequent freedom from vibration periods within the range of driving speed, and its relative accessibility.

The engine is 3⅜ in. by 5 in., block cast and of the L-head type, with a compression ratio of 4.51. It was designed to develop 85 hp. at 3000 r.p.m.

The engine parts generally are of the usual type with the exception of the crankshaft. In this the crankpins for cylinders Nos. 3, 4, 5, and 6 lie in one plane at right angles to the common plane of the crankpins for cylinders Nos. 1, 2, 7, and 8, and it is claimed that such an arrangement eliminates excessive vibration. The nine-bearing construction is used to provide a small intervening span between the main bearings giving a rigid support to the crank shaft in addition to which a vibration damper is fitted to the front end of the crankshaft. (*Automotive Industries*, vol. 48, no. 24, June 14, 1923, pp. 1266–1274, illustrated, *d*)

THE NEW SATURATION PROBLEM, Alvan Macauley. The author, who is president of the Packard Motor Car Company, discusses the physical saturation of the country with motor cars. By this he means that the number of motor cars already in existence has become so large as to make driving difficult and unpleasant in many instances.

Already traffic conditions are forcing voluntary limitations on the use of automobiles. Many suburban residents have stopped bringing their cars into the city; many city owners will not use them at all during certain hours; in the shopping and theatrical districts of New York, walking, even in bad weather, is preferred by some to the exasperating experience of trying to get to destination in a taxicab.

Above all, the problems of public safety cannot be dodged. It is not relevant to blame the "jay walker." It is not relevant to flourish statistics showing that fewer people die in automobile accidents than of measles. It is not relevant to prove that politics and the police department are responsible.

Many remedies have been suggested but they will all be merely temporary, as they do not go to the root of the difficulty.

Repetitions of New York's traffic difficulties can be avoided—and must be. The fundamental traffic problem is the problem of the city plan. Traffic crowding, delays, accidents, high taxes for police purposes and relief streets, high costs of deliveries of food—all these problems and many others can be traced in large measure to the planlessness of our cities.

A few municipal authorities and city-planning engineers are studying the unexpected increase of automobile traffic and the provisions which should be made for it. Too few public officials and public bodies realize what is needed. They do not see yet that it is cheaper to plan ahead than to cut new avenues through big buildings or rear bridges many miles long over crowded streets.

It is up to the automobile industry to face this new saturation problem frankly and intelligently, and to do all in its power to help plan and build our cities so that our industry can safely and effectively serve them. (*Automotive Industries*, vol. 48, no. 23, June 7, 1923, pp. 1231, *p.A*)

PHYSICS

THE REDUCTION OF ALL PHYSICAL DIMENSIONS TO THOSE OF SPACE AND TIME, Prof. A. P. Mathews. The author believes that ultimately all physical entities of the universe will be expressed in terms of four dimensions—three of space and that of time. With these four dimensions the whole of the physical universe could be expressed in equations of a four-dimensional space with the sole reservation that this would leave out the fifth dimension, namely, psychism, by which the author means that property of matter at present neglected by the physicists which is exhibited in its clearest form in living things and which shows itself in thought and consciousness in such large psychic units as human beings.

From this the author proceeds in an attempt to show how all the physical phenomena of the world may be expressed in dimensions of space and time. He considers for this the dimensions of quantity of electricity, dimensions of magnetism or magnetic flux, and of ethereal constants.

The reduction of physical quantities to space and time is claimed to make clear the nature of energy, which the author expresses as

$$(E) = \left(\frac{L^5}{T^2}\right) = \left(\frac{L^3}{T} \times L^2 \times \frac{1}{T}\right) = \text{magnetic flux} \times \text{electric quantity per second.}$$

In other words, all energy of whatever kind, potential or kinetic, is the product of magnetic flux and electric current, or magnetic flux times electric quantity per second, or magnetic flux times electric quantity times a frequency. This magnetic flux may be identified, as is usual, with ether flow (that is, electric quantity times velocity), and electric quantity with ether twist. Hence all energy is in the ether and is nothing else than a certain quantity of twisted ethereal flow or strain.

That potential energy is nothing else than some kind of strain in the ether, the latter being perfectly elastic, is generally supposed. So all the energy of position, whether this be gravitational energy, as in the separation of masses; electrostatic, as in the separation of charges; or chemical, as in the separation of magnetic poles or currents, is nothing else than a strain, twist, or what not in the ether. It is an ethereal phenomenon. It is always and everywhere magnetic flux times electric energy times frequency. Similarly with kinetic energy; for kinetic energy is the expression of inertia and elasticity. Inertia is self-induction. To move an electric charge from rest, or to increase its velocity when in motion, strain in the ether is set up or increased just as in separating unlike charges to make potential energy. The greater the velocity, the greater the mass; i.e., the greater the number of electric charges moved, the greater is this strain. When a moving body or current stops, or is retarded, this strain or energy or deformation is suddenly

released. Consequently all energy, whether kinetic or potential, is ethereal strain. And quantity of energy is quantity of ethereal distortion per second.

Temperature has the dimensions of energy, and is also magnetic flux times electric discharge per second. Temperature is nothing less than the product of magnetism by current.

The author further develops the above ideas by considering the dimensions of a natural physical phenomenon on the basis of an assumption that time is a four-spatial coördinate. If this is so, then mass and energy have the dimensions of volume. On the other hand, if time is kept as a separate entity not to be equated with the length, then energy is more than mass. (*Journal of the Washington Academy of Sciences*, vol. 13, no. 10, May 19, 1923, pp. 195–210, *t*)

POWER-PLANT ENGINEERING

The Yarrow Stationary Water-Tube Boiler

THE YARROW LAND-TYPE BOILER. Description of a Yarrow boiler intended for use in electric power stations, in particular the type installed at the Dunston Power Station of the Newcastle-upon-Tyne Electric Supply Co. The boiler described has been in operation for several months.

Fig. 4 illustrates the general arrangement of the boiler at Dunston, which is of the Yarrow straight-tube type, fitted with a superheater and air heaters, the elements comprising a steam drum 5 ft. in diameter and four water drums. The steam drum and the water drums are connected by $1^3/_4$-in. cold-drawn steel tubes, which are expanded and bell-mouthed at each end by pneumatic power in accordance with the usual practice in this type of boiler. The outside water drums are 30 in. in diameter and are connected to the steam drum by twelve rows of tubes, while the two drums nearer the fire, which are 23 in. in diameter, are joined to the steam drum by four rows of tubes. The total evaporating surface of the boiler is 9100 sq. ft., of which 2135 sq. ft. are provided in the two tube nests on either side of the fire and 6965 sq. ft. in the outer nests.

At each side of the boiler between the two nests of tubes one-half of the superheater is installed, the steam passing first through one and then crossing by a pipe to the other.

The superheater, which has a heating surface of 2000 sq. ft., is composed of a number of U-tubes, $1^1/_8$ in. in diameter, which are expanded and bell-mouthed into a circular drum, access to the tube ends being obtained from inside the drum—which may be entered by a manhole in the usual way. The superheater is of the self-draining type, and is designed so that the steam makes two passages through each part, or four passages in all. The drop in steam pressure through each part under the highest rates of evaporation does not exceed 1 lb. per sq. in.

Features of the boiler are the large combustion space of 1800 cu. ft. and the extent of the evaporating surface which is exposed to the direct radiation of the fire. An examination of the combustion chamber after the boiler had been in commission for some months showed that the tubes had remained quite straight.

The air heaters are placed in the uptake. They are of the tubular type and contain 1504 tubes 8 ft. in length and $2^3/_4$ in. in external diameter, the combined heating surface being 8631 sq. ft. The gases pass through the inside of the tubes, and the air to the grates is heated by passing over the outside of the tubes. The air to the grates is not only heated by passing through the air heater itself, but it also absorbs additional heat during its passage over the boiler casings, both to and from the heater. By this means the boiler casing is used to assist in heating the air and at the same time the radiation losses through the casing are reduced.

In tests the normal output of the boiler is 64,000 lb. of steam from and at 212 deg. fahr. The trials, however, were made from a feed temperature of 80 deg. fahr. to superheated steam of about 720 deg. fahr., which at the working pressure of 215 lb. per sq. in. corresponds to a degree of superheat of about 330 deg. fahr. This feedwater temperature was used on the trials and was fixed to meet certain specified conditions, but in general practice it is considerably higher. An output of 64,000 lb. of water evaporated per hour from and at 212 deg. fahr. corresponds to an evaporation of rather more than 7 lb. of water from and at 212 deg. fahr. per hour per square foot of generating surface. At this rate of evaporation, approximately 30 lb. of coal were burned per square foot of grate area per hour. An analysis of coal was obtained in accordance with recognized methods, and an efficiency on the normal load trial of 86 per cent was obtained, the analysis of the waste gases showing 12.6 per cent of CO_2. During the overload trials the output obtained was approximately 90,000 lb. of water evaporated per hour from and at 212 deg. fahr., and it was shown that the limit of output on this trial was determined by the capacity of the forced-draft fan, which had not been designed for so high an overload demand; otherwise there was no reason why the boiler should not have been worked up to a greater output.

There is one feature in the trials which is particularly interesting, namely, the difference between the draft at the funnel base and that over the grate. At normal load the draft at the funnel base was

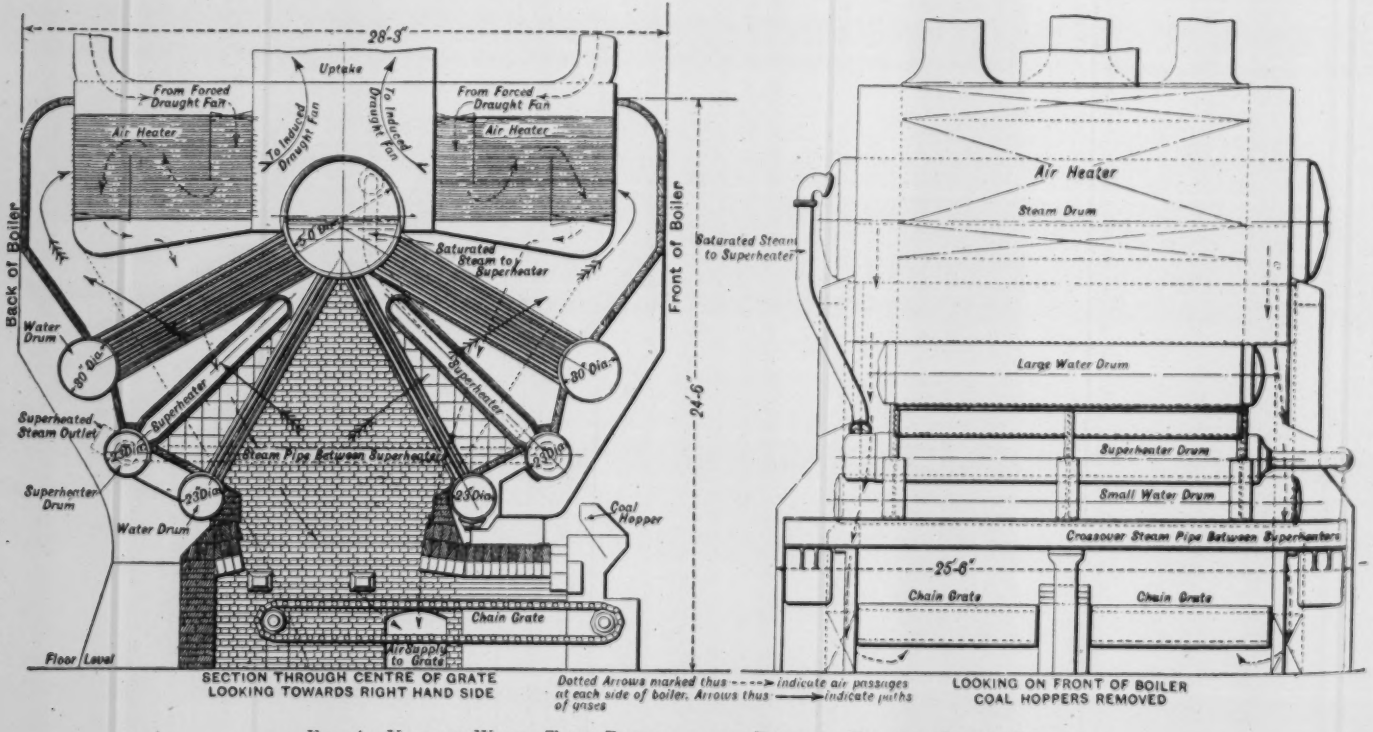

FIG. 4 YARROW WATER-TUBE BOILER AT THE DUNSTON ELECTRIC POWER STATION

0.32 in. of water, while a similar reading taken over the grate was 0.25 in. This small difference is to be attributed by the large area provided for the passage of the gases from the furnace, through the nests of tubes to the chimney, as well as to the absence of baffles or other restriction. Possibly the absence of bird-nesting experienced with this boiler is to some degree to be accounted for by the slow velocity of the gases through the tubes.

At normal load the temperature of the flue gases was reduced by 220 deg. fahr. by the air heater, while the average temperature of the air passing under the grate was about 300 deg. fahr. So far no injurious effects have been experienced on the grates or the brickwork or any other part of the boiler by air temperatures up to 362 deg. fahr., which is the limit reached during the trials, and it would appear that the grates might have been worked at a still higher temperature without harm.

The Dunston boiler was designed for coal firing, but other boilers illustrated in the original article have been arranged also for oil or powdered fuel. The original article shows also a boiler with superheater on one side, and likewise a boiler arranged with the downtake to economizer. (*The Engineer*, vol. 135, no. 3520, June 15, 1923, pp. 638–640, 7 figs., *dA*)

N.E.L.A. Report on Prime Movers

REPORT OF PRIME MOVERS COMMITTEE, NATIONAL ELECTRIC LIGHT ASSOCIATION. Only a few parts of this interesting document can be abstracted owing to lack of space.

Higher Steam Pressures and Temperatures. Considerable progress has been made toward the adoption of higher pressures and temperatures in new central-station plants since last year's report was written. A few plants have been operating with 350 lb. and 600 to 650 deg. fahr. at the boiler for several years, and these conditions have been adopted in additions now being made to existing plants.

One company, after several years' operating experience with 350 lb. at the boiler and 300 lb. at the turbine, is building a new plant using 400 lb. at the boiler. One of the large operating companies is building a plant designed for 400 lb. at the boiler and another a plant for 550 lb. In the latter case the boilers themselves can be operated up to 650 lb. The steam temperature in these later installations ranges between 650 to 750 deg. fahr.

In considering the improvement in economy which can be gained through the use of higher pressures and temperatures, the possibilities in the use of lower vacuum should always be kept in view. The temperature range worked through is of primary importance, and it may be possible under certain conditions to extend this range with a smaller increase in first cost by working at a lower vacuum than by increasing pressure and temperature. In studying this problem the relationship of the yearly load curve to the condensing-water temperature must be considered. The sub-committee on condensing equipment is studying these conditions and expects to discuss the problem in its report.

The high-pressure plants now under construction or about to be constructed represent a considerable advance in American practice. Great credit is due to the engineers and executives who have taken these forward steps in development. In a few years there will be available a considerable quantity of actual operating data on items that are, now, more or less a matter of conjecture. It is felt that in going beyond 400 lb. pressure at present, a field is being entered where little is known about the operating difficulties which may be encountered, and many engineers feel that it would be well not to advance too rapidly until more is known about these difficulties and more actual operating experience is available.

On the other hand, it unquestionably would be an error in judgment to continue installing plants designed for 200 to 250 lb. pressure at the boiler without giving due consideration to the possibilities latent in the use of higher pressures. The successful operation of 300-lb. and 350-lb. plants over a period of years shows unquestionably the wisdom of having gone to a higher pressure than the old standard.

Attention is directed to a paper presented at the 1922 Annual Meeting of The American Society of Mechanical Engineers by Geo. A. Orrok, Mem. A.S.M.E., on The Commercial Economy of High Pressure and High Superheat in the Central Stations. Mr. Orrok's article covers the effect of regeneration as well as the effect of increased temperatures and pressures.

There is quoted in full in the original report an article by L. E. Kemp which appeared in the *Electrician* of June 30, 1922. This article gives an analysis of this problem from the English viewpoint. Certain editorial comments are made on this article and additional curves are given which present Mr. Kemp's conclusions in a rather different manner.

Station Piping. Preliminary investigation of the subject of pressure drop in steam lines was started by the Committee and up to the present time a limited amount of data for specific cases has been made available by the member companies.

It would seem from some of the data submitted that considerable variation in the results obtained will occur unless measurements are made more carefully than has been the case in some instances. This is particularly true in the case of drop through reducing fittings, venturi valves, elbows, tees, and bends. Gage connections at points where direction of flow changes should be so located as to get the average pressure. The crowding of steam toward the outside of turns of all kinds is probably responsible for many discordant results. In the matter of non-return valves, checks, etc., allowance must be made for the pressure required to float the disks. Large drops through valves and fittings are not necessary. Proper design will reduce pressure losses to comparatively small amounts. The extent to which such losses can be reduced is indicated by the following tabulation which gives data for one make of throttle valve.

	Steam flow, lb. per hr.	Pressure, lb.	Temperature, deg. fahr.	Pressure drop, lb.
20-in Schutte & Koerting throttle valve built in 1915	425,000	208	541	11.0
	225,000	208	541	3.5
18-in. Schutte & Koerting valve built at a later date	320,000	212	525	1.0

The report also contains a discussion of the power-station heat balance, dealing in particular with power-station auxiliaries. This is not suitable for abstracting. Among other things this part of the report contains description of various auxiliaries, together with a brief bibliography. (*National Electric Light Association*, Report presented at the New York Convention, June, 1923, 78 pp., 87 figs., *gd*)

The Kestner Vertical Water-Tube Boiler

THE KESTNER BOILER. Description of a boiler invented by the French engineer, Paul Kestner, and built in England. It consists essentially (Fig. 5) of an upper steam drum 1, and a lower mud drum 2, which are connected together by straight vertical tubes expanded into accurately machined holes. The tubes are arranged in the form of a repeated letter V and there are only two rows.

The feed arrangement is of particular interest. The feed is delivered into a distributing tray 7, from which it overflows at definite points. It then passes down the small circulating tubes and on striking the deflector tray 8, it has its velocity reduced prior to its return to the steam drum by way of the evaporating tubes 4 and 5.

It is claimed that on account of the feed and circulating arrangements very little scaling takes place inside the boiler even when using comparatively hard water. To provide for expansion the boiler is supported by the upper or steam drum which rests on a steel staging 21, and the lower mud drum is supported by the tubes. The circulation tubes 6, are directly heated by the gases and do not produce steam.

The grate 11 is placed on a level with the mud drum 2, and in front of it; the mud drum being shielded from the hot fuel bed and the hot gases by a firebrick wall 12. Above the grate is an inclined arch 13 which is continued up to the tubes, and firebrick baffles 14 are arranged as an extension of the arch in order to fill up the gaps between the tubes. Complete combustion of the fuel takes place under the refractory brick arch, the hydrocarbons distilled off from the fuel bed by the heat of combustion being ignited and burnt under high temperature conditions before they come into contact with the tube surfaces.

To gain access to the interior of the boiler for inspection and cleaning, it is simply necessary to remove two manhole covers. Access may also be gained to the exterior portions of the tubes through a number of openings 20.

The simplicity of the Kestner boiler enables it to be opened up,

cleaned, and inspected in a very short space of time, and there is therefore a considerable reduction in the cost of labor. The ease with which the tubes can be removed and replaced will be understood from Fig. 5. The firebrick slates shown at 10 in Fig. 5 are removable, and can easily be taken off to enable the tubes to be withdrawn, the new tube being placed in position in the same manner. The length of the tubes in the standard boiler is 23 ft., but where head room is of importance the makers supply another type with shorter tubes.

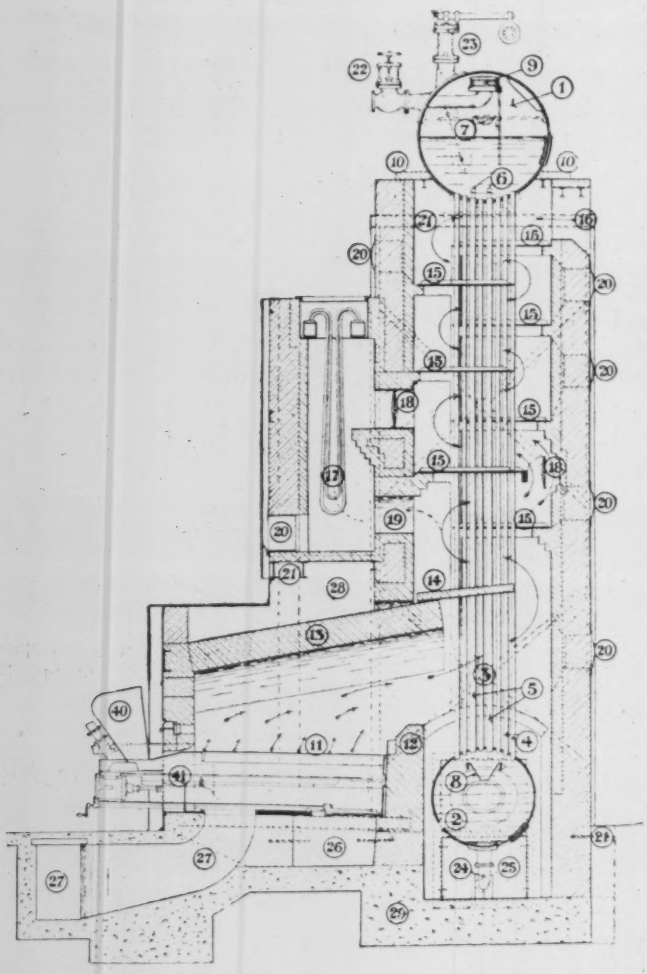

Fig. 5 The Kestner Boiler

For gas firing the Kestner boiler is claimed to be specially suitable, and, where necessary, it can be arranged for use with coal or gas, an extra combustion chamber being provided for the combustion of the latter. It is also suitable for the utilization of waste heat. All the superheater joints are accessible, and any tube expanding or plugging that may be necessary can be carried out outside the boiler setting. Steam can be raised very rapidly, and hogging of the tubes is said to be practically unknown. The boiler is claimed to have a more rapid steam circulation than any other steam generator on the market, thus allowing of a high evaporation per square foot of heating surface and a high rate of combustion of inferior fuels. The hottest gases impinge upon that part of each evaporating tube which only contains water, while the cooler gases make contact with that part of the tube which mainly contains steam. (*The Practical Engineer*, vol. 67, no. 1890, May 17, 1923, pp. 278–279, 4 figs., *d*)

Design Features of New 1200-lb.-Pressure Boilers. Description of an installation designed by the Babcock & Wilcox Co. for the Commonwealth Edison Co. of Chicago for installation at the Calumet Station. In this steam will be furnished at 1200 lb. pressure to an extra-high-pressure turbine which will exhaust at about 300 lb. into the superheater supplying steam to the steam mains feeding the main station turbine. The extra-high-pressure turbo-generator is expected to be of a capacity from 2500 to 3000 kw. The boiler will furnish about 110,000 lb. of steam per hour. The unit is of the inclined-header cross-drum type and comprises a Babcock & Wilcox forced-draft chain-grate stoker 24 ft. wide and 18 ft. 3 in. long, a lower deck of eight-high sections of 2-in. tubes having a setting height of 25 ft. 9¾ in., a primary and a secondary superheater in an interdeck space 8 ft. 1¼ in. between decks, an upper deck of seventeen-high sections of 2-in. tubes, two rows of horizontal 3½-in. circulating tubes entering a 48-in. cross-drum, and a Babcock & Wilcox steel contra-flow economizer. The lower deck is not baffled; the upper deck has a vertical baffle causing the gases to make two passes. The complete unit is about 28 ft. wide, 36½ ft. deep, including the economizer, and 45 ft. high above the floor. The heating surfaces have not yet been definitely fixed, but they will be approximately as follows: Boiler, 15,750 sq. ft.; primary superheater, 2120 sq. ft.; secondary superheater, 3300 sq. ft.; economizer, 9230 sq. ft.

One marked difference in the construction of this boiler is in the cross-drum. This drum is a forged steel cylinder, 48-in. diameter with 4-in. walls made up by rolling on a mandrel and swaging the ends.

Drum heads are integral, each with a manhole closed by a 12- by 16-in. manhole fitting. In order that the holes for connecting the circulating tubes to the drum may leave the largest practicable ligaments between circumferential rows of holes, an unusual arrangement of connections has been adopted. At the top of each uptake header two horizontal circulators are connected, but the circulators from each alternate header are bent downward and sidewise so that they are connected to the drum in the same circumferential row as the circulators without such bends. This makes the distance between these circumferential rows about 16 in. In the same way the downtake nipples are not straight but bent so that the two downtake headers are connected to each of those circumferential rows of holes in the drum.

Connections to the drum comprise four 3-in. nozzles for safety valves, a 3-in. nozzle for the saturated steam connection to the lower header of the primary superheater, and two 3-in. nozzles for connections to a feed pipe extending the length of the drum below the water line. (*Power Plant Engineering*, vol. 27, no. 12, June 15, 1923, pp. 637–638, 1 fig., *dA*)

A New Boiler-Feeding Device

The Auto-Thermal Boiler Feed. Description of a novel type of boiler-feed device having the following cycle of operations. All available condensation from steam traps brought by pipes *A*, Fig. 6, together with the necessary make-up feed by the pipe *B* is run into the upper of the two vessels *C*, which is provided with a vapor outlet

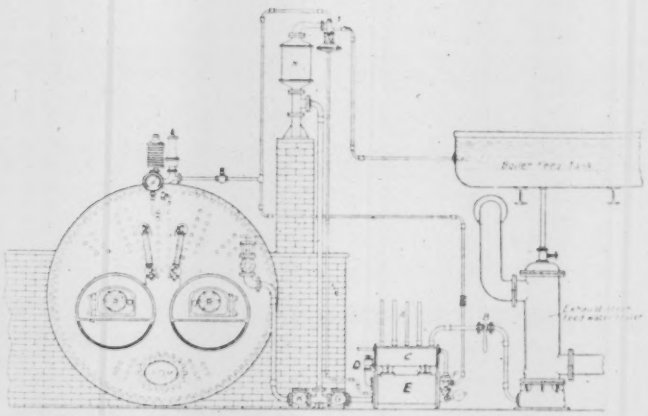

Fig. 6 Auto-Thermal Boiler Feed

and is simply a collector. From this collector the water runs by gravity through the connecting pipe *D*, which is provided with a stop valve and non-return valve, into the lower vessel *E*. An internal float rising with the water line operates the steam valve *F*, which at the psychological moment admits live steam to the vessel, discharges the water through the non-return valve *G*, and lifts it up into the vessel *H* (the feeder) above the boiler. An equilibrium

pipe connects the inside of the lifter with the underside of a diaphragm J, by means of which the rise and fall of pressure in the lifter E, in performing its functions, is utilized to operate the valves which exhaust or admit live steam to the feeder H at the proper time. The mechanism is so arranged that when the lifter is discharging under pressure the diaphragm valve is lifted, exhausting the feeder H in order to receive the water. On the lifter exhausting when empty, the valve J returns to its normal position and steam of full boiler pressure is admitted to the top of the feeder H, whence the pressure being equal to that in the boiler, the water gravitates to the boiler by way of the non-return valve K; meanwhile, the water is again filling up the lifter E and the cycle is repeated.

The perfect working of the appliance is mainly due to the effective way in which the exhausting or admission of steam to the feeder H is absolutely controlled by the lifter E. Naturally the reliability of the apparatus is also dependent upon the main steam valve F. The valve, which is moved by the steam pressure itself, is in turn controlled by a small subsidiary balanced valve. Hence the float is only called up to exert the small effort required to move the balanced valve, the rest of the work being performed by the steam pressure. (*The Iron and Coal Trades Review*, vol. 106, no. 2881, May 18, 1923, p. 747, *d*)

RAILROAD ENGINEERING

PILLIOD-SLEEVE-VALVE PERFORMANCE ON THE MONON RAILROAD. Date from an initial report on the performance of a Monon Mikado-type locomotive equipped with Pilliod valves.

This valve arrangement comprises a pair of sleeves, each actuated from a different source, which, combined with a piston moving within these sleeves, enables the exhaust and steam admission to the cylinders to be independently controlled. The valve provides for a 95 per cent release at all points of cut-off, thus eliminating excessive compression in the cylinders.

A number of indicator cards taken from the locomotive in regular operation are reproduced in the original article. These in comparison with a card taken on another locomotive of the same class under similar conditions show clearly that at the same point of cut-off the locomotive with Pilliod sleeve valves develops a higher mean effective pressure than the locomotive with ordinary valves. Consequently the Pilliod sleeve valve, in effect, enables the locomotive to develop the same tractive effort as the plain-valve locomotive with a shorter cut-off. In other words, the Pilliod sleeve valves should enable a locomotive to perform the same amount of work with less steam, used more expansively, than a locomotive equipped with an ordinary type of piston or slide valve.

The basis for economy in the use of Pilliod sleeve valves includes of course, other factors that can be more fully appreciated from a study of the thermodynamic theories involved. The theory evolved by Carnot shows that efficiency depends upon the range of temperature in the engine cylinder, and this ideal diagram shows complete expansion and compression of steam. The efficiency of the actual engine can approach this ideal but never equal it. This is due to the fact that in the use of steam in a metal cylinder practical difficulties prevent a realization of the ideal. It is apparent that after the steam is in the cylinder there is but one way to get maximum efficiency, and that is by making its terminal temperature a minimum. This indicates that release should be carried out as near the end of the stroke as possible. With the ordinary locomotive valve release does occur near the end of the stroke when there is a long cut-off but when the reverse lever is moved toward the center, thus shortening the cut-off; the release is also removed further from the end of the stroke, thus increasing compression. In the usual locomotive valve, when the engine is "cutting off" at 25 per cent of the stroke, the release has dropped back to 60 per cent, giving only 60/25 or 2.4 expansions, when under ideal conditions there should be 100/25 or 4 expansions. In this condition, compression occurs at about 40 per cent of the stroke. (*Railway Review*, vol. 72, no. 24, June 16, 1923, pp. 988–992, illustrated, *d*)

REFRIGERATION

THE COMPRESSION REFRIGERATION CYCLE, W. H. Motz. In the abstract of this article from *Refrigerating Engineering*, vol. 9, no. 9, March, 1923, which appeared on page 366 of the June issue of MECHANICAL ENGINEERING, it was stated that the amount of ammonia evaporated per minute per ton of refrigeration has been taken for standard conditions as 0.442 lb. It should have read 0.422 lb., as 0.442 lb. of ammonia per ton of refrigeration per minute would not be required unless the condensing temperature was exceedingly high or the evaporation temperature was exceedingly low.

SPECIAL PROCESSES

MAKING WELDED PRESSED-STEEL AUTOMOBILE WHEELS. Description of a process developed by the Stanley Steel Welded Wheel Corporation, Boston, Mass.

Essentially each wheel consists of a spider made up of two flanges welded together and containing eight spokes welded into pockets formed by depressions in the two flange members. Two of these spiders are butt-welded at the hub portions of adjacent flanges, so that the spokes of one spider will alternate with the spokes of the other. This unit is then assembled into the outer rims, which in their turn are provided with pockets into which the spokes are welded in the final assembling.

The actual presswork is of the usual character and it is the welding that is of particular interest. An automatic machine for spot-welding the wheels is described in the original article. The upper electrode holder is held in a frame that pivots to exert pressure on the weld, this pressure being automatically controlled by cam action. The shaft, by means of which the pressure is transmitted, carries a heavy coil spring and a hand wheel for setting the pressure. The spring is capable of transmitting 600 lb. pressure, and the arrangement for adjusting it is such that foot pressure is not required on the control. The entire cycle of operations in making a spot weld includes depressing the electrode, holding it down under pressure for a given time, switching on the current, breaking the current, and finally releasing the pressure.

After the spokes have been spot-welded through the flanges, two spiders, one outer and one inner, are butt-welded on a special machine.

Pressed-steel automobile wheels are said to weigh about the same as wood-spoked wheels, but they are much stronger. It is also claimed that they increase the life of the tire, which is believed to be due to the fact that the pressed-steel wheel possesses greater heat conductivity and as a result the temperature of the tire is reduced. (*Machinery* (N. Y.), vol. 29, no. 10, June, 1923, pp. 757–760, 8 figs., *d*)

TESTING AND MEASUREMENT

ACOUSTIC METHOD OF FINDING ELASTIC LIMIT OF METALS. According to experiments conducted by the physics department of Lehigh University, Bethlehem, Pa., it is believed that stretching a metal until it squeaks and listening for the squeak with a microphone will give a test showing how much strain that the metal can stand when used in a place of stress, such as a bridge girder or railroad rail. Those conducting the experiments point out that the tests may result in a novel and quick method for determining the elastic limit of metals. Forgings could be tested rapidly before they are put in use.

Several metals have been used in the tests. While the investigators listened through a sensitive microphone similar to the dictaphone for the faintest sound, the metal was subjected to a gradually increasing pull until it broke. It was found that squeaking, rasping sounds were produced after the pull had reached a certain definite value, and this value was different for the various metals used. The tests indicated that the sounds were not produced until the elastic limit of the material had been reached. The work will be continued. (*Iron Trade*, vol. 72, no. 23, June 7, 1923, p. 1671, *e*)

CLASSIFICATION OF ARTICLES

Articles appearing in the Survey are classified as *c* comparative; *d* descriptive; *e* experimental; *g* general, *h* historical; *m* mathematical; *p* practical; *s* statistical; *t* theoretical. Articles of especial merit are rated *A* by the reviewer. Opinions expressed are those of the reviewer, not of the Society.

Standard Screw Threads for Bolts, Machine Screws, Nuts, and Commercially Tapped Holes

First Report of Sectional Committee on the Standardization and Unification of Screw Threads

THE long-looked-for report of the Sectional Committee on the Standardization and Unification of Screw Threads is now before the Councils of the Society of Automotive Engineers and The American Society of Mechanical Engineers. Upon approval and adoption by these boards it will be submitted by them to the American Engineering Standards Committee for formal adoption as an American Standard.

This Sectional Committee was organized under the Procedure of the A.E.S.C. in the fall of 1920, the S.A.E. and the A.S.M.E. acting as Joint Sponsors. The Committee's instructions called for a complete review of the Progress Report of the National Screw Thread Commission, which was then being printed. In April, 1921, the members of the Committee began their detailed study of the report and on June 9, 1921, they met for the formal organization of the Committee and the election of officers. Luther D. Burlingame was elected Chairman, Ralph E. Flanders, Vice-Chairman, and Clifford B. LePage, Secretary. Realizing the difficulty of a committee of 25 functioning efficiently in the preparation of a report of this kind, a Working Committee of seven members of the Sectional Committee was immediately appointed. This Working Committee consisted of Ellwood Burdsall, Chairman, Paul W. Abbott, Earle Buckingham, George S. Case, Ralph E. Flanders, George E. Hammann, and Will R. Porter.

After a very careful study of the N.S.T.C. report and numerous meetings, the Working Committee issued its first report to the members of the Sectional Committee in February, 1922. Since that date four additional revised drafts of this report have been prepared and carefully considered by the Sectional Committee. As the report neared its final form the Sectional Committee delegated Messrs. Flanders, Buckingham, Case, and Wells as a special committee to discuss with the National Screw Thread Commission the slight modifications which the Sectional Committee desired to make. The resulting discussions in the meetings of the Commission led to the adoption of modifications in the report which were mutually satisfactory.

Purpose of Present Report

During its early discussion of the report of the Working Committee the Sectional Committee definitely limited the field to be covered by its first report to Screw Threads for Bolts, Machine Screws, Nuts, and Commercially Tapped Holes. A number of considerations prompted this action. In the first place it was desired to put in the hands of American industry promptly one of the tools which it greatly needs, viz., a manual of improved screw-thread practice. In the second place it found that standards for pipe threads, fire-hose coupling threads, and small hose threads had either been passed through the A.E.S.C. procedure or were on the way by other means, so consideration of them was omitted. But one part of the Commission's report therefore remains to be accounted for. This is Section VI, entitled Gages, and Appendix 6, Gages and Methods of Test. By a vote of the Sectional Committee it was decided not to recommend any one particular system of gages or for gaging screw threads but to prepare and issue as a second report a general discussion on this subject in which certain fundamentals only will be laid down.

Critical Comment

The first report of the Sectional Committee on the Standardization and Unification of Screw Threads is merely a selection from and a rearrangement of the material which was originally

Table 1 The Two-Thread Series

1	2	3	4
Identification	Basic Major Diameter (Inches)	Fine Series Threads per Inch	Coarse Series Threads per Inch
0	0.0600	80	..
1	0.0730	72	64
2	0.0860	64	56
3	0.0990	56	48
4	0.1120	48	40
5	0.1250	44	40
6	0.1380	40	32
8	0.1640	36	32
10	0.1900	32	24
12	0.2160	28	24
1/4	0.2500	28	20
5/16	0.3125	24	18
3/8	0.3750	24	16
7/16	0.4375	20	14
1/2	0.5000	20	13
9/16	0.5625	18	12
5/8	0.6250	18	11
3/4	0.7500	16	10
7/8	0.8750	14	9
1	1.0000	14	8
1 1/8	1.1250	12	7
1 1/4	1.2500	12	7
1 1/2	1.5000	12	6
1 3/4	1.7500	12[1]	5
2	2.0000	12[1]	4 1/2
2 1/4	2.2500	12[1]	4 1/2
2 1/2	2.5000	12[1]	4
2 3/4	2.7500	12[1]	4
3	3.0000	10[1]	4

[1] These pitches are suitable for work requiring short engagement where it is not practical to specify tolerances for all conditions. They are not, therefore, included in the succeeding tables.

Table 8 Coarse Thread Series—Free Fit (Class 2)—Screws

1	2	3	4	5	6	7	8	9
		Major Diameter			Pitch Diameter			
Size	Threads per Inch	Maximum[1] (Inches)	Tolerance (Inches)	Minimum (Inches)	Maximum[2] (Inches)	Tolerance[3] (Inches)	Minimum (Inches)	Maximum[2] Minor Diameter (Inches)
1	64	0.0730	0.0038	0.0692	0.0629	0.0019	0.0610	0.0538
2	56	0.0860	0.0040	0.0820	0.0744	0.0020	0.0724	0.0641
3	48	0.0990	0.0044	0.0946	0.0855	0.0022	0.0833	0.0754
4	40	0.1120	0.0048	0.1072	0.0958	0.0024	0.0934	0.0813
5	40	0.1250	0.0048	0.1202	0.1088	0.0024	0.1064	0.0943
6	32	0.1380	0.0054	0.1326	0.1177	0.0027	0.1150	0.0997
8	32	0.1640	0.0054	0.1586	0.1437	0.0027	0.1410	0.1257
10	24	0.1900	0.0066	0.1834	0.1629	0.0033	0.1596	0.1389
12	24	0.2160	0.0066	0.2094	0.1889	0.0033	0.1856	0.1649
1/4	20	0.2500	0.0072	0.2428	0.2175	0.0036	0.2139	0.1887
5/16	18	0.3125	0.0082	0.3043	0.2764	0.0041	0.2723	0.2443
3/8	16	0.3750	0.0090	0.3660	0.3344	0.0045	0.3299	0.2983
7/16	14	0.4375	0.0098	0.4277	0.3911	0.0049	0.3862	0.3499
1/2	13	0.5000	0.0104	0.4896	0.4500	0.0052	0.4448	0.4056
9/16	12	0.5625	0.0112	0.5513	0.5084	0.0056	0.5028	0.4603
5/8	11	0.6250	0.0118	0.6132	0.5660	0.0059	0.5601	0.5135
3/4	10	0.7500	0.0128	0.7372	0.6850	0.0064	0.6786	0.6273
7/8	9	0.8750	0.0140	0.8610	0.8028	0.0070	0.7958	0.7387
1	8	1.0000	0.0152	0.9848	0.9188	0.0076	0.9112	0.8466
1 1/8	7	1.1250	0.0170	1.1080	1.0322	0.0085	1.0287	0.9497
1 1/4	7	1.2500	0.0170	1.2330	1.1572	0.0085	1.1487	1.0747
1 1/2	6	1.5000	0.0202	1.4798	1.3917	0.0101	1.3816	1.2955
1 3/4	5	1.7500	0.0232	1.7268	1.6201	0.0116	1.6085	1.5046
2	4 1/2	2.0000	0.0254	1.9746	1.8557	0.0127	1.8430	1.7274
2 1/4	4 1/2	2.2500	0.0254	2.2246	2.1057	0.0127	2.0930	1.9774
2 1/2	4	2.5000	0.0280	2.4720	2.3376	0.0140	2.3236	2.1933
2 3/4	4	2.7500	0.0280	2.7220	2.5876	0.0140	2.5736	2.4433
3	4	3.0000	0.0280	2.9720	2.8376	0.0140	2.8236	2.6933

[1] Basic diameter.

[2] Dimensions given are figured to intersection of the worn tool arc with a centerline through crest and root. Minimum flat at root equals $1/8 \times p$.

[3] The tolerances specified for pitch diameter are cumulative and include all errors of lead and angle.

published in the Progress Report of the National Screw Thread Commission. Several "paper" changes have been made by the Committee but these did not alter materially the product itself. Any threaded product which meets the specifications laid down by the Progress Report of the National Screw Thread Commission will also meet the specifications of the Sectional Committee's report.

This report does not present a new set of standards for screw threads but rather an extension of existing standards by specifying limiting dimensions for the various sizes and pitches of threads already in common use. It also gives official sanction to certain processes such as oversize drill sizes which have long been established by common use. Altogether it is an attempt to codify the best current practice on screw-thread production. How well this attempt has succeeded can be best learned by its general use in industry.

The Committee's decision to limit the field covered by its first report to threaded parts for holding purposes, such as bolts, nuts, machine screws, and commercially tapped holes, seems to be well founded. When the standard practice codified by this report has had a fair tryout by the American screw-thread industry, given an opportunity for revision or correction if necessary, a long advance will have been made toward the further establishment of standards for threading tools and screw threads for special purposes. Among the latter are included studs, fine threads of large diameter and short engagement, for instance, those which are used for cap nuts on machine tool collars, hub and radiator caps, etc. The Sectional Committee fully realized that the screw-thread field is so broad as to make it impossible to cover the entire field in its first report. Furthermore the experience which it is hoped to gain by the use of its first proposals will be of great value in working out satisfactory standards for other applications.

Since it is impossible for us to reproduce in MECHANICAL ENGINEERING the complete report of the Sectional Committee, we are printing several sections of this report, including four tables to indicate the complete and altogether satisfactory way in which this Committee now presents the results of its two years' study. The report is now being edited for publication in pamphlet form and it is expected that these pamphlets will be ready for general distribution by August 15. Those desiring copies should apply to The American Society of Mechanical Engineers.

II Form of Thread

The form of thread profile as specified herein, and known previously as the United States Standard or Sellers' Profile, is adopted and shall hereafter be known as the American (National) Form of Thread.

1 Where Used. The American (National) Form of Thread Profile shall be used wherever possible for screw-thread work.

2 Specifications. The basic angle of thread between the sides of the thread measured in an axial plane shall be 60 degrees. The line bisecting this 60 degree angle shall be perpendicular to the axis of the screw thread.

The basic width of flat at the root and crest of the thread form is found as follows:

$$F = {}^1/_8 \times p \text{ or } 0.125 \times p$$

The basic depth of the thread form is found as follows:

$$h = 0.649519 \times p = \frac{0.649519}{n}$$

where p = pitch in inches

n = number of threads per inch.

3 Clearance in Nut at Minor Diameter. A clearance shall be provided at the minor diameter of the nut by removing the thread form at the crest by an amount between one-sixth and one-fourth of the basic thread depth.

4 Clearance in Nut at Major Diameter. A clearance at the major diameter of the nut shall be provided by decreasing the depth of the truncation triangle any desired amount down to one-third of its theoretical value.

TABLE 9 COARSE THREAD SERIES—FREE FIT (CLASS 2)—NUTS

1	2	3	6	7	8	9	10	11
Size	Threads per Inch	Minimum[1,2] Major Diameter (Inches)	Pitch Diameter			Minor Diameter		
			Minimum[1] (Inches)	Tolerance[3] (Inches)	Maximum (Inches)	Minimum (Inches)	Tolerance (Inches)	Maximum (Inches)
1	64	0.0730	0.0629	0.0019	0.0648	0.0561	0.0017	0.0578
2	56	0.0860	0.0744	0.0020	0.0764	0.0667	0.0019	0.0686
3	48	0.0990	0.0855	0.0022	0.0877	0.0764	0.0023	0.0787
4	40	0.1120	0.0958	0.0024	0.0982	0.0849	0.0027	0.0876
5	40	0.1250	0.1088	0.0024	0.1112	0.0979	0.0027	0.1006
6	32	0.1380	0.1177	0.0027	0.1204	0.1042	0.0034	0.1076
8	32	0.1640	0.1437	0.0027	0.1464	0.1302	0.0034	0.1336
10	24	0.1900	0.1629	0.0033	0.1662	0.1449	0.0045	0.1494
12	24	0.2160	0.1889	0.0033	0.1922	0.1709	0.0045	0.1754
1/4	20	0.2500	0.2175	0.0036	0.2211	0.1959	0.0054	0.2013
5/16	18	0.3125	0.2764	0.0041	0.2805	0.2524	0.0060	0.2584
3/8	16	0.3750	0.3344	0.0045	0.3389	0.3073	0.0068	0.3141
7/16	14	0.4375	0.3911	0.0049	0.3960	0.3602	0.0077	0.3679
1/2	13	0.5000	0.4500	0.0052	0.4552	0.4167	0.0084	0.4251
9/16	12	0.5625	0.5084	0.0056	0.5140	0.4723	0.0090	0.4813
5/8	11	0.6250	0.5660	0.0059	0.5719	0.5266	0.0098	0.5364
3/4	10	0.7500	0.6850	0.0064	0.6914	0.6417	0.0109	0.6526
7/8	9	0.8750	0.8028	0.0070	0.8098	0.7547	0.0120	0.7667
1	8	1.0000	0.9188	0.0076	0.9264	0.8647	0.0135	0.8782
1 1/8	7	1.1250	1.0322	0.0085	1.0407	0.9704	0.0154	0.9858
1 1/4	7	1.2500	1.1572	0.0085	1.1657	1.0954	0.0154	1.1108
1 1/2	6	1.5000	1.3917	0.0101	1.4018	1.3196	0.0180	1.3376
1 3/4	5	1.7500	1.6201	0.0116	1.6317	1.5335	0.0216	1.5551
2	4 1/2	2.0000	1.8557	0.0127	1.8684	1.7594	0.0241	1.7835
2 1/4	4 1/2	2.2500	2.1057	0.0127	2.1184	2.0094	0.0241	2.0335
2 1/2	4	2.5000	2.3376	0.0140	2.3516	2.2294	0.0270	2.2564
2 3/4	4	2.7500	2.5876	0.0140	2.6016	2.4794	0.0270	2.5064
3	4	3.0000	2.8376	0.0140	2.8516	2.7294	0.0270	2.7564

[1] Basic diameter.

[2] Dimensions given are allowable only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage by interference with rounded roots due to worn tap. Minimum flat at root equals $1/24 \times p$.

[3] The tolerances specified for pitch diameter are cumulative and include all errors of lead and angle.

TABLE 17 FINE THREAD SERIES—FREE FIT (CLASS 2)—NUTS

1	2	3	6	7	8	9	10	11
Size	Threads per Inch	Minimum[1] Major Diameter[2] (Inches)	Pitch Diameter			Minor Diameter		
			Minimum[1] (Inches)	Tolerance[3] (Inches)	Maximum (Inches)	Minimum (Inches)	Tolerance (Inches)	Maximum (Inches)
0	80	0.0600	0.0519	0.0017	0.0536	0.0465	0.0013	0.0478
1	72	0.0730	0.0640	0.0018	0.0658	0.0580	0.0015	0.0595
2	64	0.0860	0.0759	0.0019	0.0778	0.0691	0.0017	0.0708
3	56	0.0990	0.0874	0.0020	0.0894	0.0797	0.0019	0.0816
4	48	0.1120	0.0985	0.0022	0.1007	0.0894	0.0023	0.0917
5	44	0.1250	0.1102	0.0023	0.1125	0.1004	0.0025	0.1029
6	40	0.1380	0.1218	0.0024	0.1242	0.1109	0.0027	0.1136
8	36	0.1640	0.1460	0.0025	0.1485	0.1339	0.0030	0.1369
10	32	0.1900	0.1697	0.0027	0.1724	0.1562	0.0034	0.1596
12	28	0.2160	0.1928	0.0031	0.1959	0.1773	0.0039	0.1812
1/4	28	0.2500	0.2268	0.0031	0.2299	0.2113	0.0039	0.2152
5/16	24	0.3125	0.2854	0.0033	0.2887	0.2674	0.0045	0.2719
3/8	24	0.3750	0.3479	0.0033	0.3512	0.3299	0.0045	0.3344
7/16	20	0.4375	0.4050	0.0036	0.4086	0.3834	0.0054	0.3888
1/2	20	0.5000	0.4675	0.0036	0.4711	0.4459	0.0054	0.4513
9/16	18	0.5625	0.5264	0.0041	0.5305	0.5024	0.0060	0.5084
5/8	18	0.6250	0.5889	0.0041	0.5930	0.5649	0.0060	0.5709
3/4	16	0.7500	0.7094	0.0045	0.7139	0.6823	0.0068	0.6891
7/8	14	0.8750	0.8286	0.0049	0.8335	0.7977	0.0077	0.8054
1	14	1.0000	0.9536	0.0049	0.9585	0.9227	0.0077	0.9304
1 1/8	12	1.1250	1.0709	0.0056	1.0765	1.0348	0.0090	1.0438
1 1/4	12	1.2500	1.1959	0.0056	1.2015	1.1598	0.0090	1.1688
1 1/2	12	1.5000	1.4459	0.0056	1.4515	1.4098	0.0090	1.4188

[1] Basic diameter.

[2] Dimensions given are allowable only with tap having theoretically sharp corners. Threaded hole must not reject correct basic "Go" gage by interference with rounded roots due to worn tap. Minimum flat at root equals $1/24 \times p$.

[3] The tolerances specified for pitch diameter are cumulative and include all errors of load and angle.

III Thread Series Adopted

Two thread series are adopted,—a coarse and a fine. The Coarse-Thread Series is the present "United States Standard" supplemented in the sizes below one-fourth inch by the standard established by The American Society of Mechanical Engineers (A.S.M.E.) in 1907. The Fine-Thread Series ($\frac{1}{4}$ inch to $1\frac{1}{2}$ inches inclusive) is in accordance with the present "Regular Screw-Thread Series of the S.A.E. Standard for Screw-Threads" established by The Society of Automotive Engineers (S.A.E.) in 1911, supplemented in sizes below one-fourth inch by the Fine-Thread Series established by The American Society of Mechanical Engineers (A.S.M.E.).

These two series are specified in Table 1.

IV Classification of Fits

1 Description

There are established herein for general use four distinct classes of screw-thread fits as specified in the following brief outline. The examples given under each class of fit are for the purpose of illustration only.

Loose Fit (Class 1) Recommended as a commercial standard for tapped holes in the numbered sizes only. May be used with screws of other classes to obtain quality of fit desired. (See Appendix C.)

Free Fit (Class 2) Includes the great bulk of screw-thread work of ordinary quality of finished and semi-finished bolts and nuts, etc. (Called "Medium Fit Regular" in the N.S.T.C. "Progress Report.")

Medium Fit (Class 3) Includes the better grade of interchangeable screw-thread work, uch ass automobile bolts and nuts (Called "Medium Fit Special" in N.S.T.C. "Progress Report").

Close Fit (Class 4) Includes screw-thread work requiring a fine snug fit, somewhat closer than the medium fit, such as high grade aircraft parts, etc. In this class of fit selective assembly of parts may be required. It is not considered practicable as a commercial standard for tapped holes of the numbered sizes.

2 A Second Method of Obtaining Medium Fit (Class 3)

A detailed examination of the tolerances given in Tables 3, 4 and 5 will indicate that the same quality of fit as given by Medium Fit screws and threaded holes can be obtained by using screws to Close Fit tolerances in holes made to Free Fit tolerances, or vice versa. The general adoption of this practice will reduce the variety of taps, gages, threaded parts, etc., required to be carried in stock. However, the combination of Free and Close Fits should be used in place of Medium Fit only when the substitution is understood and agreed to by all parties concerned.

4 General Specifications

The following general specifications will apply to all classes of fits specified in the body of the report. For special specifications applying to the Loose Fit class see Appendix C.

(a) Uniform Minimum Nut. The pitch diameter of the minimum threaded hole or nut corresponds to the basic size, variations being permitted above the basic size. The major and minor diameters of the minimum nut are also uniform for all classes of fit.

(b) Uniform Tap Drill Sizes. The maximum and minimum minor diameters and the consequent minor diameter tolerances are the same for all nuts of a given size for all classes of fit. This permits uniform tap drill sizes for all classes of fit.

(c) Uniform Major and Minor Diameter of Screws. The maximum and minimum major and minor diameters are the same for all screws of a given size for all classes of fit herein tabulated.

(d) Length of Engagement. The tolerances herein determined are based on a length of engagement not to exceed the nominal diameter of the thread. Where greater lengths of engagement are required a corresponding increase in the accuracy of lead and thread form is necessary, which may be obtained by the provision of selection of more accurate threading tools, and the use of longer "Go" gages.

(e) Rounded Root Forms. The crest clearances allowed are such as to permit rounded root forms in both nut and screw. These may be formed either by tools purposely rounded or rounded as a result of wear in use. For the limits of this permissible rounding see Pages 28 and 29, Par. 6 and Par. 7.

Roller Chains and Sprockets

Report of the Joint Committee Which is Composed of Representatives of the S.A.E., the A.G.M.A., and the A.S.M.E.

THE organization of a roller-chain committee was approved by the Council of The American Society of Mechanical Engineers at the Spring Meeting of the Society in 1917. The Committee was accordingly organized at Indianapolis, Ind., September 14, 1917. Its second meeting and all subsequent meetings have been held in conjunction with the Roller Chain Division of the Society of Automotive Engineers. At a later date the Sprocket Committee of the American Gear Manufacturers Association was invited to coöperate in the standardization work of the Joint Committee. The Committee will welcome suggestions for corrections or additions to this report. These should be addressed to the Secretary, Geo. M. Bartlett, in care of the Society, 29 W. 39th St., N. Y. City.

Recommendations of the Committee, relating to the standardization of chains and sprockets which have been approved by the Council of the A.S.M.E. and published in the September, 1921, issue of MECHANICAL ENGINEERING, are as follows:

1 Series of Pitches
2 Roller Diameters
3 Pin Diameters
4 Thickness of Side Plates
5 Widths of Narrow and Wide Series
6 Heavy, Medium, and Light Series
7 Names of Chain Parts
8 Tolerance for Chain Length
9 Measuring Load
10 Test Load
11 Uniform System of Numbering Chains
12 Thickness and Chamfer of Sprocket Teeth
13 Standard Tooth Form
14 Range of Teeth for Each Cutter
15 Number of Teeth on Which to Base Each Cutter
16 Sprocket-Cutter Outlines
17 Outside Diameters for Sprockets
18 Thickness of Cutters.

NEW RECOMMENDATIONS

The following proposed standards have been approved at meetings of the Joint Roller Chain Committee and are now before the Council of The American Society of Mechanical Engineers for approval and adoption.

19 The *thickness of center plates* for double, triple, and quadruple chains shall be equal to two thicknesses of the regular inside plates as used in the single standard chain.

20 The *minimum breaking strength* of standard chains, when used for publication, shall be $105{,}000 \times (\text{Pin Diam.})^2 - 700$ lb.

21 The following note is to be attached to the formula for minimum breaking strengths.

NOTE: These are the loads at which the chains may break or become permanently stretched. The actual *working* loads may range from $\frac{1}{5}$th to $\frac{1}{20}$th of the ultimate strength, depending upon the speed, teeth in wheels, conditions of lubrication, and pulsations arising from either power or load end. Chains should not be selected on the basis of breaking strengths.

22 Revise the *standard measuring load* from "1 per cent of the *average* breaking strength" to 1.25 per cent of the *minimum* breaking strength. Reason: The minimum breaking strength is given by a formula—the average breaking strength is not.

23 Revise the *standard test load* from "$\frac{1}{3}$ of the *average* breaking strength" to 40 per cent of the *minimum* breaking strength. Reason: same as for 22.

24 Revise the standard nomenclature with respect to the terms "assembled pins," "pin link," and "connecting link" to agree with that given below under the heading Roller-Chain Nomenclature.

25 Four *types of sprocket cutters* to be recognized, namely:

Space Cutters, of which five will be required to cut from 7 teeth up for any given roller diameter. The ranges to be respectively 7–8 teeth, 9–11 teeth, 12–17 teeth, 18–34 teeth, 35 teeth and over. The use of less than 7 teeth is discouraged, but when necessary single-purpose cutters of this type are to be used.

Straddle Cutters, of which two will be required to cut from 7 teeth up for any given pitch and roller diameter. Cutter B is recommended for 17 teeth and under (or for more than 17 teeth if a low pressure angle is desired). Cutter A is recommended for 18 teeth

and over (or for less than 18 teeth if a large pressure angle is desired and the arc of contact between chain and sprocket is fairly large).

Hobs, of which only one will be required to cut any number of teeth for a given pitch and roller diameter.

Fellows Cutters, for use on the Fellows gear shaper, of which only one will be required to cut any number of teeth for a given pitch and roller diameter.

26 All *cutters to be marked* giving pitch, roller diameter, and range of teeth to be cut.

27 *Sprocket Cutter Bores* (recommended practice) to be approximately:

$$0.7\sqrt{\text{Width of cutter} + \text{roller diam.} + 0.7P}$$

where P is the pitch. See Table 8.

28 *Minimum Outside Diameters of Space Cutters* for 35 teeth and over (recommended practice) to be approximately:

$$1.2\,(\text{Bore} + \text{roller diam.} + 0.7P) + 1 \text{ in.}$$

See Table 1.

29 Minimum outside diameters of space cutters for less than 35 teeth (recommended practice) to be as per Table 1.

30 *Minimum Outside Diameters of Straddle Cutters* (recommended practice) to be same as for space cutters for 35 teeth and over. See Table 4.

31 *Size of Keyways* in cutters to be in accordance with present prevailing cutter-arbor practice as given in Table 8.

32 *Number of Cutting Teeth in Sprocket Cutters* to be left to the individual cutter makers.

33 The *standard tooth form for block and twin-roller chains* shall follow the same essential specifications as for standard roller chains with the same angle of bend.

34 Recommended *widths of cutters* as per Table 8.

35 Recommended *designations* for the four principal types of sprockets are as follows:

Type A—plain plate
Type B—single hub
Type C—double hub
Type D—detachable hub

36 *Revised Sprocket-Tooth Form.* The Joint Committee now submits herewith a proposed modification of the standard sprocket-tooth form as passed upon by the Technical Sub-Committee, and approved at the meeting of the Joint Committee held in New York, January 8, 1923.

In the original design very careful consideration was given to the working arcs of the teeth. The exact height of the tooth was set in an empirical manner, as it was not an important operating element. Subsequent developments have indicated that the long tooth restricted the range of teeth that could be covered by a given cutter, whereas it seemed desirable to increase this range for certain special cases. This was accomplished by shortening the tooth and giving greater clearance between the roller and the point of the tooth.

Also the probability of an increased use of hobs and Fellows cutters for generating the teeth has increased the desirability of modifying the angle of tooth gap for larger numbers of teeth so as to conform more closely to the generating process and at the same time to produce a desired increase in pressure angle for the larger sprockets.

Roller-Chain Nomenclature

Roller Link. An inside link consisting of two inside plates, two bushings, and two rollers.

Pin Link. An outside link consisting of two pin-link plates assembled with two pins.

Inside Plate. One of the plates forming the tension members of a roller link.

Pin-Link Plate. One of the plates forming the tension members of a pin link.

Pin. A stud articulating within a bushing of an inside link and secured at its ends by the pin-link plates.

Bushing. A cylindrical bearing in which the pin turns.

Roller. A ring or thimble which turns over a bushing.

Assembled Pins. Two pins assembled with *one* pin-link plate.

Connecting Link. A pin link with one side plate detachable.

Connecting-Link Plate. The detachable pin-link plate belonging to a connecting link.

Offset Link. A link consisting of two offset side plates assembled with a bushing and roller at one end and an offset-link pin at the other.

Offset-Link Pin. A pin used in offset links.

Design of Standard Sprocket Tooth

P = pitch; D = nominal roller diameter; T = number of teeth

$D' = 1.005D + 0.003$ in.; $A = 35^\circ + \frac{60}{T}$; $A' = 35^\circ - \frac{120}{T}$;

$B = 18^\circ - \frac{56}{T}$;

$ac = 0.8D$; $M = 0.8D\cos\left(35^\circ + \frac{60}{T}\right)$;

$N = 0.8D\sin\left(35^\circ + \frac{60}{T}\right)$

$J = 0.8D\sin\left(35^\circ - \frac{120}{T}\right)$; $K = 0.8D\cos\left(35^\circ - \frac{T}{120}\right)$

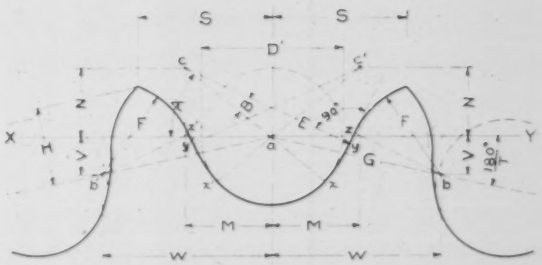

Fig. 1 Tooth Form

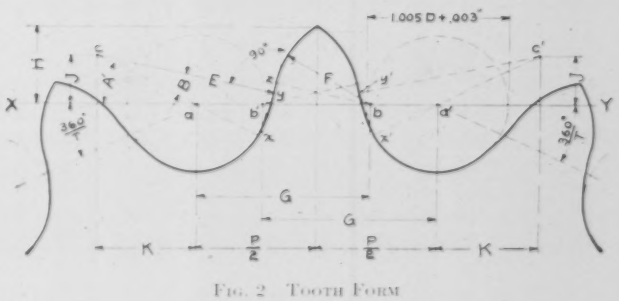

Fig. 2 Tooth Form

$E = 1.3025D + 0.0015$ in.

Chord $xy = (2.605D + 0.003)\sin\left(9^\circ - \frac{28}{T}\right)$

$yz = D\left[1.24\sin\left(17^\circ - \frac{64}{T}\right) - 0.8\sin\left(18^\circ - \frac{56}{T}\right)\right]$

$G = 1.24D$; $W = 1.24D\cos\frac{180}{T}$; $V = 1.24D\sin\frac{180}{T}$

$F = D\left[0.8\cos\left(18^\circ - \frac{56}{T}\right) + 1.24\cos\left(17^\circ - \frac{64}{T}\right) - 1.3025\right] - 0.0015$ in.

$H = \sqrt{F^2 - \left(1.24D - \frac{P}{2}\right)^2}$. When $1.24D$ is less than $\frac{P}{2}$, then $H = F$

$S = \frac{P}{2}\cos\frac{180}{T} + H\sin\frac{180}{T}$

Outside diameter of sprocket when tooth is pointed $= P\cot\frac{180}{T} + 2H$

Minimum outside diameter of blank $= P\left(0.6 + \cot\frac{180}{T}\right)$

The pressure angle for a new chain is $xab = 35^\circ - \frac{120}{T}$

The minimum pressure angle is $xab - B = 17^\circ - \frac{64}{T}$

The average pressure angle is $26^\circ - \frac{92}{T}$

STANDARD "SPACE" CUTTERS FOR SPROCKET TEETH

Space cutters are made for the following ranges of teeth: 7–8; 9–11; 12–17; 18–34; 35 and over. The lowest number of teeth in any group is designated by n, and the highest by N.

The cutters are based upon an intermediate number of teeth, T, equal to $2Nn \div (N + n)$; but the topping curve zs is based upon N teeth. The values of T for the several cutters are 7.47; 9.9; 14.07; 23.54; and 56.

Space cutters designed for a given roller diameter D will cut sprockets of any pitch.

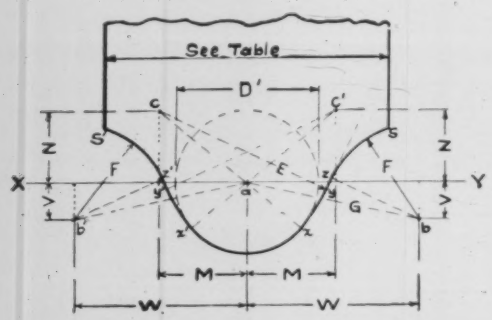

FIG. 3 SPACE CUTTER

Construction. Referring to Fig. 3, draw XY. With a as center and a radius ax equal to $\frac{1}{2}D'$ draw the circular arc for the seating curve xx' (Table 2). Locate c and c' from dimensions M and N (Table 1). With c and c' as centers describe the arcs yx and $x'y'$. Draw yz perpendicular to cy. Locate b from dimensions W and V, draw bz parallel to cy, and with radius bz, equal to F (Table 1), draw the topping curve zs. The line yz is a common tangent to the two circular arcs xy and zs.

Supplementary Data. Tables 1, 2, and 3 give all necessary data for the construction of standard space cutters. Supplementary formulas used in their calculation are given below.

The angle Yab is $180°/T$ when the cutter is made for T teeth only; but it has values as given in Table 3 for cutters covering a range of teeth as here designed. Accordingly the following formulas are special for cutters covering the standard ranges of teeth:

$W = 1.24D \cos Ya\ b$; $V = 1.24D \sin Yab$

$$bc' = b'c = \sqrt{(W - M)^2 + (V + N)^2}$$

$$yz = D\left[1.24 \sin\left(17° + \frac{116}{T} - Yab\right) - 0.8 \sin\left(18° - \frac{56}{T}\right)\right]$$

$$F = D\left[0.8 \cos\left(18° - \frac{56}{T}\right) + 1.24 \cos\left(17° + \frac{116}{T} - Yab\right) - 1.3025\right] - 0.0015 \text{ in.}$$

$$cb = c'b' = \sqrt{(E + F)^2 + yz^2}$$

Width of cutter $= 1.02P\left(0.6 + \cot\frac{180}{n}\right)\sin\frac{180}{n}$ to the next higher 32nd inch. Where the same roller diameter is commonly used with chains of two different pitches it is recommended that stock cutters be made wide enough to cut sprockets for both chains.

TABLE 1 DATA FOR LAYING OUT CUTTER OUTLINES

Range of teeth	M	N	W	V	F	Chord xy	yz
7–8	$0.585D$	$0.5457D$	$1.1327D$	$0.504D$	$0.7104D$ $-0.0015''$	$0.2384D$ $+0.0003''$	$0.039D$
9–11	$0.599D$	$0.5303D$	$1.1782D$	$0.387D$	$0.6981D$ $-0.0015''$	$0.28D$ $+0.0003''$	$0.056D$
12–17	$0.619D$	$0.5068D$	$1.2128D$	$0.258D$	$0.6807D$ $-0.0015''$	$0.3181D$ $+0.0004''$	$0.090D$
18–34	$0.634D$	$0.4879D$	$1.2353D$	$0.108D$	$0.6542D$ $-0.0015''$	$0.354D$ $+0.0004''$	$0.146D$
35 up	$0.647D$	$0.471D$	$1.24D$	0	$0.6345D$ $-0.0015''$	$0.385D$ $+0.0004''$	$0.171D$

TABLE 2 SEATING-CURVE DIAMETERS

Roller diam. D, in.	Seating-curve diam. D', in.	Roller diam. D, in.	Seating-curve diam. D', in.
0.200	0.204	3/4	0.757
1/4	0.254	7/8	0.882
0.306 & 5/16	0.317	1	1.008
0.400	0.405	1 1/8	1.134
15/32	0.474	1.55	1.561
9/16	0.568	1 9/16	1.574
5/8	0.631	...	...

TABLE 3 CHECKS

Teeth	E	G	$\frac{b-c}{b'-c'}$	$\frac{b-c'}{b'-c}$	Angle Yab
7–8	$1.3025D + 0.0015$	$1.24D$	$2.0133D$	$1.1841D$	24°
9–11	$1.3025D + 0.0015$	$1.24D$	$2.0014D$	$1.0847D$	18°10′
12–17	$1.3025D + 0.0015$	$1.24D$	$1.9852D$	$0.9682D$	12°
18–34	$1.3025D + 0.0015$	$1.24D$	$1.9621D$	$0.8465D$	5°
35 up	$1.3025D + 0.0015$	$1.24D$	$1.9445D$	$0.7576D$	0°

DESIGN OF STANDARD STRADDLE CUTTERS

Only two cutters of this type are required to cover the entire range of teeth.

P = pitch; D = roller diameter; T = number of teeth on which cutter is based.

Construction. Draw XY and the two seating-curve circles as shown in Fig. 4, making radius $ax = 0.5025D + 0.0015$ in.

Locate c and c' from the dimensions K and J as given in Table 4. Locate b and b'. Draw cax and $c'a'x'$ and with centers c and c' draw the "working curves" xy and $x'y'$. Draw yz and $y'z'$ perpendicular to cy and $c'y'$, respectively. Draw bz and $b'z'$ parallel to cy and $c'y'$, respectively. With b and b' as centers and radius equal to bz, strike the arcs of the "topping curves." The dimensions given in Table 4 may be calculated from the formulas given in connection with Fig. 2.

TABLE 4 DATA FOR LAYING OUT STRADDLE-CUTTER OUTLINES

Cutter	To cut	Based on	K	J	F	Chord xy	yz	E	Pressure Angle Avg.	Pressure Angle Max.
B	$17T$ and under	11 Teeth	$0.730D$	$0.327D$	$0.6937D$ -0.0005	$0.2928D$ $+0.0003$	$0.0617D$	$1.3025D$ $+0.0015$	17.6°	24.1°
A	$18T$ and over.	40 Teeth	$0.678D$	$0.424D$	$0.6596D$ -0.0015	$0.3726D$ $+0.0004$	$0.1007D$	$1.3025D$ $+0.0015$	23.7°	32°

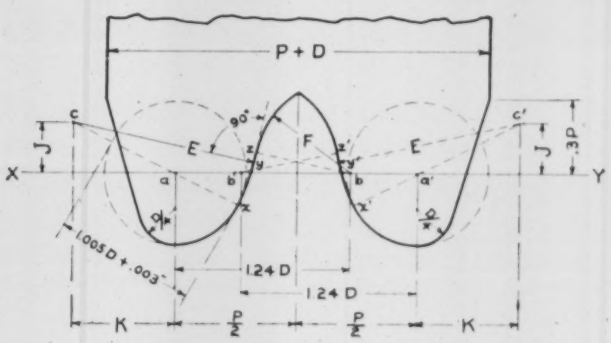

FIG. 4 STRADDLE CUTTER

The maximum pressure angle is $xab = 35° - \frac{120}{T}$

The minimum pressure angle is $xab - acy = 17° - \frac{64}{T}$

The average pressure angle is $26° - \frac{92}{T}$

STANDARD SPROCKET-TOOTH FORM FOR BLOCK CHAINS

L = pitch of blocks; e = pitch of side plates; P = nominal pitch of chain $= L + e$; D = diameter of round end of block; T = number of teeth.

$L' = L + 0.005D + 0.003$ in.; $A = 35° + \frac{120}{T}$; $B = 18° - \frac{28}{T}$

$ac = 0.8P$; $M = 0.8D \cos\left(35° + \frac{120}{T}\right)$;

$N = 0.8D \sin\left(35° + \frac{120}{T}\right)$

$E = 1.3025D + 0.0015$ in.

Chord $xy = 2.6D \sin\left(9° - \frac{14}{T}\right)$

$$yz = D\left[1.24\sin\left(17^\circ - \frac{32}{T}\right) - 0.8\sin\left(18^\circ - \frac{28}{T}\right)\right]$$

$$G = 1.24D;\quad W = 1.24D\cos\frac{180}{T};\quad V = 1.24D\sin\frac{180}{T}$$

$$F = D\left[0.8\cos\left(18^\circ - \frac{28}{T}\right) + 1.24\cos\left(17^\circ - \frac{32}{T}\right) - 1.3\right]$$

$$H = \sqrt{F^2 - \left(1.24D - \frac{P - L'}{2}\right)^2}, \text{ very nearly,}$$

$$S = H\sin\frac{180}{T} + \frac{P - L'}{2}\cos\frac{180}{T} + \frac{L'}{2}, \text{ very nearly.}$$

Outside diameter of sprocket when tooth is pointed =

$$2H + \frac{L' + (P - L')\cos\frac{180}{T}}{\sin\frac{180}{T}}, \text{ very nearly.}$$

$$\text{Minimum outside diameter of blank} = 0.6c + \frac{L + e\cos\frac{180}{T}}{\sin\frac{180}{T}}$$

The pressure angle for a new chain is $xab = 35^\circ - \frac{60}{T}$

The minimum pressure angle is $xab - B = 17^\circ - \frac{32}{T}$

The average pressure angle is $26^\circ - \frac{46}{T}$

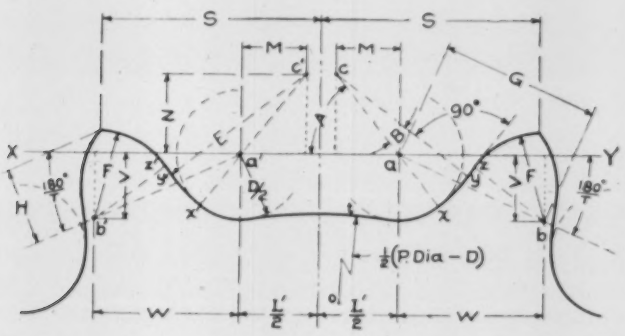

FIG. 5 TOOTH FORM FOR BLOCK CHAINS

STANDARD CUTTERS FOR BLOCK-CHAIN SPROCKETS

Sprocket cutters for block and twin-roller chains are made for the same ranges of teeth as for roller-chain sprockets and are based upon the same number of teeth.

Construction. Referring to Fig. 6, draw XY. Locate a and a' making L' = pitch of block + $0.005D$ + 0.003 in., and with these points as centers (radius = $\frac{1}{2}D$) draw the circular arcs for the seating curves. With radius ot, equal to one-half the bottom diameter of the largest sprocket to be cut, draw an arc tangent to the two seating curves. Locate c and c' from dimensions M and N (Table 5). With c and c' as centers describe the arcs xy and $x'y'$. Draw yz and $y'z'$ perpendicular to cy and $c'y'$, respectively.

TABLE 5 DATA FOR LAYING OUT BLOCK-CUTTER OUTLINE

Range of teeth	M	N	W	V	F	Chord xy	yz	ot	Q
6	0.4589D	0.6554D	1.0738D	0.620D	0.6927D	0.3019D	0.0662D	0.97P − 0.5D	1.12P
7–8	0.5027D	0.6223D	1.1327D	0.504D	0.6845D	0.3227D	0.0779D	1.28P − 0.5D	1.12P
9–11	0.5348D	0.5950D	1.1782D	0.387D	0.6764D	0.3432D	0.0861D	1.76P − 0.5D	1.11P
12–17	0.5801D	0.5510D	1.2128D	0.258D	0.6638D	0.3622D	0.1111D	2.71P − 0.5D	1.10P
18–34	0.6120D	0.5152D	1.2353D	0.108D	0.6431D	0.3801D	0.1577D	5.41P − 0.5D	1.08P
35 up	0.6403D	0.4796D	1.24D	0	0.6308D	0.3955D	0.1763D	x	1.05P

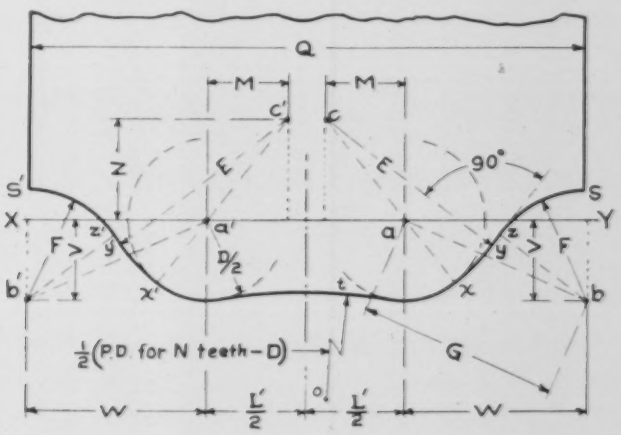

FIG. 6 BLOCK CUTTER

Locate b and b' from dimensions W and V. With these as centers and a radius F, draw the arcs ZS and $Z'S'$. The line yz is a common tangent to the curves xy and zs.

Supplementary Data. Formulas used in the calculations of data in Tables 6 and 7 are given below.

The angle Yab is $180°/T$ when the cutter is made for T teeth only, but it has values as given in Table 6 for cutters covering a range of teeth as here designed. Accordingly the following formu-

(Continued on page 511)

TABLE 6 CHECKS

Range of teeth	E	G	Angle Yab
6	1.3D	1.24D	30°
7–8	1.3D	1.24D	24°
9–11	1.3D	1.24D	18°10′
12–17	1.3D	1.24D	12°
18–34	1.3D	1.24D	5°
35 up	1.3D	1.24D	0°

TABLE 7 RECOMMENDED SIZES OF CUTTERS FOR BLOCK-CHAIN SPROCKETS

Pitch of Chain	Block End Diam., in.	Pitch of Block, in.	Cutter Diam., in.	Bore, in.	Keyway, in.	Cutter Width 6T	7–8T	9–11T	12–17T	18–34T	35T up
1	0.325	0.400	3	1	3/32 × 3/64	13/16	13/32	13/32	11/8	11/8	19/32
1½	0.531	0.563	4	1¼	3/16 × 3/32	125/32	13/4	13/4	119/16	119/16	19/8

TABLE 8 RECOMMENDED SPROCKET-CUTTER SIZES

Pitch, in.	Roll Diam., in.	Cutter diameter (min.), in. 6T	7–8T	9–11T	12–17T	18–34T	35T and over	Cutter width (min.), in. 6T	7–8T	9–11T	12–17T	18–34T	35T and over	Bore	Keyway
*3/8	0.200	23/4	23/4	23/4	23/4	23/4	23/4	15/32	15/32	15/32	7/16	7/16	13/32	1	3/32 × 3/64
3/8	0.250	27/8	27/8	27/8	27/8	27/8	27/8	15/32	15/32	15/32	7/16	7/16	13/32	1	3/32 × 3/64
3/8 to 1/2	0.250	23/4	27/8	27/8	27/8	27/8	27/8	4/8	5/8	19/32	19/32	9/16	9/16	1	3/32 × 3/64
1/2	0.313	27/8	3	3	3	3	3	5/8	5/8	19/32	19/32	9/16	9/16	1	3/32 × 3/64
*1/2 to 5/8	0.313	3	3	31/8	31/8	31/8	31/8	5/8	5/8	5/8	23/32	11/16	11/16	1	3/32 × 3/64
*5/8	0.400	31/8	31/8	31/4	31/4	31/4	31/4	5/8	5/8	5/8	23/32	11/16	11/16	1	3/32 × 3/64
5/8 to 3/4	0.400	31/4	31/4	33/8	33/8	33/8	33/8	29/32	29/32	29/32	7/8	27/32	13/16	1	3/32 × 3/64
*3/4	0.469	31/4	31/4	33/8	33/8	33/8	33/8	29/32	29/32	29/32	7/8	27/32	13/16	1	3/32 × 3/64
3/4 to 1	0.469	31/4	33/8	33/8	31/2	31/2	31/2	11/4	11/4	13/16	15/16	11/8	13/32	1	3/32 × 3/64
*1	0.563	33/4	37/8	37/8	4	4	4	11/4	11/4	13/16	15/16	11/8	13/32	11/4	3/16 × 3/32
1	0.625	37/8	4	4	41/8	41/8	41/8	11/4	11/4	13/16	15/16	11/8	13/32	11/4	3/16 × 3/32
*1 to 1¼	0.625	37/8	4	41/8	41/8	41/4	41/4	11/2	11/2	119/32	115/32	113/32	113/32	11/4	3/16 × 3/32
1¼	0.750	4	41/8	41/4	41/4	43/8	43/8	11/2	11/2	119/32	125/32	113/32	111/32	11/4	3/16 × 3/32
*1¼ to 1½	0.750	41/4	43/8	41/2	41/2	43/8	43/8	113/16	113/16	125/32	15/4	111/16	15/8	11/4	3/16 × 3/32
*1½	0.875	43/4	47/8	5	5	51/8	51/8	113/16	113/16	125/32	13/4	111/16	13/8	11/2	5/16 × 6/32
1½ to 1¾	0.875	47/8	5	51/8	51/4	53/8	53/8	23/32	23/32	21/16	21/32	131/32	17/8	11/2	5/16 × 5/32
*1¾	1.000	5	51/8	51/4	53/8	51/2	51/2	23/32	23/32	21/16	21/32	131/32	17/8	11/2	5/16 × 5/32
1¾ to 2	1.000	51/8	51/4	53/8	51/2	55/8	55/8	213/32	213/32	23/4	25/16	21/4	25/32	11/2	5/16 × 5/32
*2	1.125	55/8	51/2	53/8	53/4	57/8	57/8	213/32	213/32	23/4	25/16	21/4	25/32	11/2	5/16 × 6/32
2 to 2½	1.125	51/2	53/4	57/8	6	61/8	61/4	3	3	215/16	229/32	25/4	211/16	11/2	5/16 × 6/32
*2½	1.550	63/8	63/8	63/4	67/8	7	71/8	3	3	215/16	229/32	25/4	211/16	15/4	3/8 × 3/16
2½ to 3	1.550	63/4	7	71/8	71/4	71/4	71/2	319/32	319/32	317/32	315/32	311/32	37/32	13/4	3/8 × 3/16
*3	1.900	71/2	73/4	77/8	8	8	81/4	319/32	319/32	317/32	315/32	311/32	37/32	2	1/2 × 1/4

* Cutter will fill all requirements for chains as made in the United States at the present time.

ENGINEERING RESEARCH

A Department Conducted by the Research Committee of the A.S.M.E.

Résumé of the Month

A—Research Results

The purpose of this section of Engineering Research is to give the origin of research information which has been completed, to give a résumé of research results with formulas or curves where such may be readily given, and to report results of non-extensive researches which in the opinion of the investigators do not warrant a paper.

Aeronautics A2-23. Aeronautic Instruments. This paper describes in detail various aeronautic instruments and will be useful to those interested in the general aspects of the aeronautic-instrument art. It was prepared by F. L. Hunt and is known as Bureau of Standards Technologic Paper No. 237. For a copy address Superintendent of Documents, Washington, D. C.; price, 20 cents.

Building Materials A2-23. Loading Test of a Hollow-Tile and Reinforced-Concrete Floors. The Arlington Building, Washington, D. C. was originally intended for a hotel with a live load of 75 lb. per sq. ft. On account of the purchase by the Government for other uses, the construction was strengthened in view of possible loads on floor panels of 100 lb. per sq. ft. The width of the panels of this building varied from about 11 to 15 ft. and the ratio of length to width varied from 1 to 2.

A 1:2:4 gravel concrete was used; that in the columns and beam stems was very "sloppy;" in the slabs it was somewhat drier. Tests on the cylindrical specimens and the difficulty of chipping the slabs in the preparation for the test showed the concrete to be of good quality. The age of the specimens was about two and one-half months. Round and square smooth bars and square corrugated bars were used for reinforcement. The tiles used in the floors appeared to be of medium-burned grade, except some of the 4-inch tiles, which apparently were hard-burned. The columns consist of a steel H-section surrounded by a thick casing of concrete, the total column being 16 in. square. Vertical rods for reinforcement were placed at each corner and held in place by square ties made of $\frac{1}{4}$-in. round rods placed 12 in. on centers and 2 in. from the surface of the concrete, which were designed to prevent spalling in case of fire.

This type of construction was new in many respects and the structure is considerably lighter than recommended by the final report (1916) of the Joint Committee on Concrete and Reinforced Concrete or allowed by most building codes. The primary purpose of the test was, therefore, to find how much live load could actually be placed on individual panels before serious stresses were developed. It was desired by all concerned in this test to secure as much information as possible on the action of the hollow-tile slabs and incidentally also on the framework when subjected to load.

The tests herein described are among the first to be made on this type of construction, which is a framework of light-weight structural-steel members used as reinforcement for the concrete columns and beams, supporting two-way reinforced concrete and terra-cotta, hollow-tile floor panels.

The authors of this Bureau of Standards Technologic Paper No. 236 are Messrs. L. J. Larson and S. N. Petrenko. The paper includes a bibliography of 29 items describing similar field tests of floors in reinforced-concrete buildings. Address Superintendent of Documents, Washington, D. C. Price, 15 cents.

Fuels A5-23. Combustion of Powdered Coal. This paper contains a review of some tests conducted by the U. S. Bureau of Mines and the Combustion Engineering Corporation on boilers fired with powdered coal, which show that coal when powdered may be burned with greater thermal efficiency for steam raising than when burned in any other way. It was presented before the section of Gas and Fuel Chemistry at the 64th meeting of the American Chemical Society, Pittsburgh, Pa. by Henry Kreisinger, Research Engineer, Combustion Engineering Corporation, and John Blizard, Fuel Engineer of the U. S. Bureau of Mines. Address H. Foster Bain, Director, U. S. Bureau of Mines, Washington, D. C. requesting copy of Serial No. 2470.

Lubricants A2-23. Physical and Chemical Properties of Lubricating Oils Made from California Crudes. This is the second of a series of reports on a survey of petroleum products manufactured on the Pacific Coast from California crude oils. California crudes differ in many characteristics from the petroleum produced in other parts of the United States, and the claim has been made that some of the present Federal Specifications for petroleum products tend to discriminate against California products. The purposes of the present survey are to determine what particular specification requirements California petroleum products characteristically fail to meet, how widely the products deviate from these requirements, and if possible to show whether the fault lies with the specifications or the products, or along what lines investigation to determine the question should proceed. The first report of the series gave the physical and chemical properties of gasoline and mineral spirits made from California crudes.

Data on the physical and chemical properties of the samples tested are given in tables, each of which is preceded by the classification and specifications for the particular class of oil considered. Following each table brief notes show wherein samples deviate from specification requirements.

Most of the oils meet the specification requirements as regards flash point, viscosity, pour point; corrosion and reaction tests, except for deviations which for the most part are of minor importance.

Those desiring full information concerning these tests should address the authors, Messrs. E. C. Lane and N. F. LeJeune at the Bureau of Mines, Washington, D. C., requesting a copy of Serial No. 2482.

Mining A1-23. Monel Metal a Material for Flame Safety Lamp Gauzes. This pamphlet known as Serial No. 2468 of the Bureau of Mines was prepared by Messrs. A. B. Hooker and R. A. Kearns.

Monel metal, the natural alloy of nickel and copper, is practically noncorrodible. Its melting point is approximately the same as that of steel. From time to time it has been suggested that this alloy might be a better gauze material than steel, and the investigation showed that it is a very satisfactory material for the purpose.

Monel-metal gauzes maintained their shape and stiffness under the greatest heat to which they were subjected in gaseous mixtures equally as well as steel gauzes did; this together with the fact they do not corrode perceptibly in humid atmospheres should assure a much longer life for Monel-metal gauzes. For copies address H. Foster Bain, Director, U. S. Bureau of Mines, Washington, D. C.

Non-Ferrous Metals A5-23. Monel-Metal a Material for Flame Safety Lamp Gauzes. See *Mining A1-23.*

Properties of Engineering Materials A1-23. Loading Test of a Hollow-Tile and Reinforced-Concrete Floors. See *Building Materials A2-23.*

Refrigeration A2-23. New Mollier Chart for Ammonia. In reference "Refrigeration A2-23" (July issue of Mechanical Engineering) the publication of Tables of Thermodynamic Properties of Ammonia was announced. These data are now available in the form of a Mollier chart.

Pressures and volumes can easily be read on this chart with an accuracy of 1 per cent and heat content to within 0.5 B.t.u. It may be used as a complete substitute for the tables in calculations which do not require the highest degree of accuracy and such computations can be carried out by this means with much greater speed than is possible from data presented in tabular form.

It is known as Bureau of Standards Miscellaneous Publication No. 52, and can be obtained from the Superintendent of Documents, Washington, D. C., at 5 cents per copy.

Safety Devices A1-23. Monel Metal a Material for Flame Safety Lamp Gauzes. See *Mining A1-23.*

B—Research in Progress

The purpose of this section of Engineering Research is to bring together those who are working on the same problem for cooperation or conference, to prevent unnecessary duplication of work, and to inform the profession of the investigations who are engaged upon research problems. The addresses of these investigators are given for the purpose of correspondence.

Automotive Vehicles and Equipment B1-23. Growth of Cast-Iron Pistons. See *Iron and Steel B2-23.*

Building Materials B2-23. Colorless Waterproofing Materials for Stone. Investigations are now under way at the Bureau of Standards covering the action of frost on building stone and on the value of colorless waterproofing materials with which to protect the surface of stone structures. During the last month the series of exposure tests on such materials, having for its object the determination of the relative durability of these treatments under weather conditions, has been supplemented by a series of tests to determine the efficiency of the different waterproofing materials in preventing decay of the stone.

Crystallization tests are being made on waterproofed specimens of stone to secure a comparison between treated and untreated specimens. Waterproofed specimens have also been exposed to the weather and will be tested after a considerable period of exposure.

Electrolysis B1-23. Effects of Impurities in Storage-Battery Electrolytes. The importance of obtaining information concerning the action of impurities in storage-battery electrolytes arises from the detrimental effects which many of them produce on the operating characteristics and life of the storage battery. Such information is necessary as a basis for the preparation of specifications covering sulphuric acid for use in batteries. In the present investigation the effects of small

amounts of iron, manganese, platinum, and copper have been determined. It was found that the presence of 1 part in 10,000,000 of platinum in the electrolyte increases the local action at the negative plates 50 per cent; the effect of copper is much less, while the effect of iron is of unusual interest because of its accelerating action at the negative plates. Manganese deposits upon the positive plates in the form of manganese dioxide which covers the active material, closes the pores, and causes a large amount of charging current to be wasted as gas. Work is being extended which will include the effect of other impurities.

Hardness B1-23. STUDY OF QUENCHING MEDIA. The object of a new investigation started by the Bureau of Standards during the past month is to obtain more complete information than is now available on the cooling properties of the ordinary quenching liquids used in the heat treatment of steel. It is also hoped that some information bearing on the subject of dimensional changes, occurring during the hardening of steel, will also be obtained. A few preliminary experiments have been carried out in quenching nickel cylinders in water with very satisfactory results. As the preliminary work is now practically completed, it is hoped that considerable progress will be made on the main investigation during the next few weeks.

Iron and Steel B2-23. GROWTH OF CAST-IRON PISTONS. This research has been conducted with special reference to the growth of the cast iron used in automobile pistons. While it has been known for years that pistons can be grown a very small amount, this investigation has shown that this amount can be made many times greater under the proper treatment, and as such developed to the point where it can be used as a certain and practical repair process.

Further work remains to be done on this subject which may be done very profitably: (1) The determination of the variation of the constituents in the material with the number of treatments, the temperature used, etc. (2) The variation of the strength under the same conditions. (3) The more complete control of warp. (4) The extent, the location and magnitude of internal strains. (5) The variation of hardness of the pistons before and after different heat treatment. This investigation has been conducted by A. R. Nottingham, State College of Washington, Pullman, Washington.

Materials B1-23. NON-DESTRUCTIVE TESTING OF WIRE ROPE. See *Properties of Engineering Materials B1-23.*

Non-Ferrous Metals B3-23. SCRATCH HARDNESS OF COPPER. The results obtained in cold rolling very pure, electrolytic, unmelted copper and also ordinary commercial copper indicate clearly that the early stages of the deformation of the metal harden it very rapidly. The maximum degree of hardness is soon reached, however, and a reversal occurs, the metal becoming very appreciably softer as the deformation process is continued, so that the metal in its final condition after cold rolling is completed, is softer as measured by the scratch hardness method than in its initial state. These results are confirmed by tests made on the same material by the micro-Brinell method. Annealing cold-rolled sheets at a low temperature (100 deg. cent.) softens the material appreciably, but the general form of the hardness deformation curve is not affected.

After annealing at a higher temperature all specimens showed the same hardness, thus indicating that thinness of the rolled material does not affect materially the results of the scratch hardness test. This is an important point and suggests the use of this method of hardness testing on such specimens as are too thin to permit the determination of hardness by other more familiar methods. The results which were obtained with pure iron and with an iron-carbon alloy (low-carbon) were of the same general character as those obtained with copper and thus helped to confirm the conclusions which had been tentatively reached.

This work which is being carried on by the Bureau of Standards will be extended to a few other metals to determine whether this behavior is peculiar to copper or more general in its nature.

Properties of Engineering Materials B1-23. NON-DESTRUCTIVE TESTING OF WIRE ROPE. An investigation of the possibilities of using non-destructive methods for testing wire rope with special reference to hoisting rope is about to be undertaken by the Bureau of Standards. Present methods of inspection are unsatisfactory and do not unfailingly tell when it is necessary to remove the rope from service. The object of this investigation is to develop, if possible, some method by which an actual test can be made to determine the conditions of the rope with respect to its deterioration in service, without destroying it. An advisory committee is being organized which will be composed of representatives of the principal interests connected with the manufacture, use, and testing of wire rope.

Steel—Its Treatment and Products B2-23. HEAT TREATMENT AND PROPERTIES OF MAGNET STEELS. The Metallurgical and Electrical Divisions of the Bureau of Standards in coöperation, are beginning an investigation to determine the effects of various heat treatments upon the characteristic properties of the steels now used commercially or suggested for use in the manufacture of permanent magnets. During the past month about 10 samples were hardened, and it is hoped that they will soon be ready for the first magnetic tests.

Safety Devices B1-23. NON-DESTRUCTIVE TESTING OF WIRE ROPE. See *Properties of Engineering Materials B1-23.*

A.S.M.E. Boiler Code Committee Work

THE Boiler Code Committee meets monthly for the purpose of considering communications relative to the Boiler Code. Any one desiring information as to the application of the Code is requested to communicate with the Secretary of the Committee, Mr. C. W. Obert, 29 West 39th. St., New York, N. Y.

The procedure of the Committee in handling the cases is as follows: All inquiries must be in written form before they are accepted for consideration. Copies are sent by the Secretary of the Committee to all of the members of the Committee. The interpretation, in the form of a reply is then prepared by the Committee and passed upon at a regular meeting of the Committee. This interpretation is later submitted to the Council of the Society, for approval, after which it is issued to the inquirer and simultaneously published in MECHANICAL ENGINEERING.

Below are given the interpretations of the Committee in Cases Nos. 414, 416 and 418, as formulated at the meeting of April 26, 1923, and Cases Nos. 419 and 420 as formulated at the meeting of May 31, 1923, and approved by the Council. In accordance with the Committee's practice, the names of inquirers have been omitted.

CASE NO. 414

Inquiry: Is it permissible, in the case of the flue of an internally-fired boiler which is corrugated for about half of its length and the remainder plain, to calculate the thickness of the plain portion in accordance with Par. 239, assuming that the length of the plain portion is the distance from the center line of the rivets in the head to the beginning of the corrugations?

Reply: It is the opinion of the Committee that in the calculation of the thickness of the plain portion of such a furnace, Par. 239*b* should be used, taking for the length the distance from the face of the flange to the center of the first definite corrugation, and the thickness thus obtained shall be uniform throughout the entire length of the flue.

CASE NO. 416

(In the hands of the Committee)

CASE NO. 418

Inquiry: Is it permissible under the requirements of the Heating Boiler Section of the Code, to attach the safety valve to the pipe-header outlet connection of a type of cast-iron sectional heating boiler with sections designed too narrow to admit of a separate opening on any one of them for the safety valve?

Reply: If the design of the boiler is such that it is not possible to form the safety-valve openings directly on the boiler sections, it is the opinion of the Committee that the sections should be connected at two or more points to a manifold or header which will serve as a steam drum, and that it will be permissible to mount the safety valve on this manifold or header.

CASE NO. 419

(In the hands of the Committee)

CASE NO. 420

Inquiry: Is it permissible, under the requirements of the Boiler Code, to drive up a blister or blisters on a tube or tubes of a water-tube boiler and reinforce them by depositing metal over the same by the autogenous process; or later when other blisters appear above or below the first reinforcement to again use the same process, increasing the length of the reinforcement; also to weld tubes lengthwise by the same process regardless of minimum or maximum length of weld for any thickness of tube and for any pressure without removing the tube from boiler or annealing the welding?

Reply: It is the opinion of the Committee that such reinforcement or welding of blisters or other defects in the tubes of water-tube boilers would be analogous to the welding of cracks in shell plates subject to tensile strain which is prohibited under the Recommendations for Repairs by Welding in the Appendix of the 1918 Edition of the Code, page 127, third paragraph.

MECHANICAL ENGINEERING

A Monthly Journal Containing a Review of Progress and attainments in Mechanical Engineering and Related Fields, The Engineering Index (of current engineering literature), together with a Summary of the Activities, Papers and Proceedings of

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An American Screw-Thread Series

LUTHER D. BURLINGAME

IT MIGHT be assumed because we have the long-established U. S. Standard for Screw Threads, and for small diameters the more recently adopted A.S.M.E. Standard, also the S.A.E. Standard for fine threads, that all the requirements for American Screw-Thread Standardization have been met. The problem of interchangeability, however, is only partly solved by the establishment of the general form of thread, and of the relation of pitch to diameter. That this is so is shown by the wide variation in screw and nut products now on the market, also of taps and dies. In spite of the degree of standardization so far attained, the screw-thread product of one manufacturer often will not interchange with that of another, because each of them is working within his own range of limits and tolerances.

The requirements of the Navy will illustrate the need of further steps in standardization. The importance of naval vessels' being able to secure from a wide variety of sources necessary supplies involving questions of screw threads is apparent, and yet at the present time interchangeability in such products cannot be assured even when they are manufactured ostensibly according to the same system.

It is to remedy this condition that further efforts at standardization have been undertaken.

As far back as 1912 the A.S.M.E. appointed a Committee on Limits and Tolerances in Screw-Thread Fits. This Committee after a number of years' work rendered a report which will be found in the A.S.M.E. Transactions, Vol. 42, page 1053, and which laid the foundation for further progress.

Members of this original Committee, recognizing that the whole screw-thread problem was too great to solve at a single effort and with limited support, took steps to secure Government coöperation, and through their influence a commission was appointed by the U. S. Government to make a thorough study of the whole matter.

The able and thorough work of this commission resulted in the publication in 1921 of the Progress Report of the National Screw Thread Commission, covering coarse and fine systems of screw threads for general use, pipe threads, hose-coupling threads, and also gaging systems, such as to insure the products' coming within the required tolerances for the classes of work for which each was intended.

This report was officially brought to the attention of the two engineering societies having the most vital interest in its findings, and they (the S.A.E. and the A.S.M.E.) were made sponsors, under the procedure of the A.E.S.C., for the purpose of reviewing the Report of the Commission.

The work of the joint committee appointed under this authorization has been done with the special purpose of adapting the report of the commission to the practical and everyday uses of industry. For that reason its work has been largely a "boiling down" and rearranging process, and the suggesting of such modifications as would make the original report more workable.

Two years of effort, including painstaking committee and subcommittee work and conferences with the National Screw-Thread Commission, have resulted in bringing about full agreement with the commission regarding suggested changes, and the Joint Committee now presents its first Report, several sections of which will be found on page 498 of this issue of MECHANICAL ENGINEERING.

The question of gaging has not been made a part of this first Report but will take the form of a later one, the subject being treated so as to lay down general principles rather than to recommend the adoption of a particular plan of gaging, as it is felt that the primary purpose is to establish the requirements of interchangeability for the product and leave to each manufacturer the determining of methods to bring it within those requirements.

While the standard here submitted represents distinctly an American standard and is the outgrowth of American practice, there is still a hope on the part of the committee that it will prove a steppingstone toward uniformity with British standards. This possibility has been in the minds of both the commission and the joint committee, resulting in incorporating in the Report such features as would most nearly accord with and aid in interchangeability with the Whitworth Standard. Study has also been given to the question as to what mutual concessions might be practicable so as to eventually bring the two systems of the English-speaking world into uniformity.

The "ice was broken" at the time of the National Screw-Thread Commission's visit to England, in 1919, when the question of the desirability and practicability of unifying the two systems was frankly discussed. The fact that they are so nearly alike as to be in most cases capable of "emergency interchangeability" gives hope for eventually bringing them together.

At the recent Montreal Meeting of the A.S.M.E. a further conference was held between the members of the joint committee and the Canadian Engineering Standards Association, and through the good offices of the latter the matter is now being further discussed with the British Engineering Standards Association in London.

If such conferences should result in bringing about uniformity in screw-thread products throughout the English-speaking world it would be a great achievement and one worthy of mutual concessions for its attainment.

Pending such further negotiations and the final approval of the sponsor bodies it is thought best to make this report now available for use.

LUTHER D. BURLINGAME[1]

Formulas for Computing Economies of Labor-Saving Equipment

The publication of the report of the Formula Committee, of the A.S.M.E. Materials Handling Division, which was presented at the Spring Meeting of the Society held in Montreal, Canada, May 28–31, 1923, has been deferred to a later issue of MECHANICAL ENGINEERING.

[1] Indus. Supt. Brown and Sharpe Mfg., Co., Providence, R. I. Chairman of Sectional Committee on Standardization and Unification of Screw Threads.

John R. Freeman Recommended for A.S.M.E. Medal for 1923

THE A.S.M.E. Committee on Awards and Prizes has recommended to the Council that the Society Medal for some notable invention or striking improvement in connection with the industries be awarded[1] for the year 1923 to Past-President John R. Freeman for his eminent service to engineering and manufacturing by his meritorious work in fire prevention and the preservation of property. The following is a brief statement of his activities in the development of the system of rules and inspection under the Factory Mutual Insurance Companies, which have saved the industries millions of dollars.

Mr. Freeman became associated with the Factory Mutual Insurance Companies in 1886 when they were small and their membership was confined mainly to New England factories. Immunity from loss was dependent mainly upon careful management and oversight rather than upon appliances for extinguishing fire. One of his earliest activities was the conducting of a series of experiments on fire streams, during which the friction loss of water flowing through various kinds of hose was determined and the most efficient shape for nozzles developed. The results of these tests, made at Lawrence in 1888, were later tabulated and published as the Fire Stream Tables. These tables are the standard used by all engineers having to do with the flow of water through nozzles or hydrants.

Harris and Ewing

JOHN R. FREEMAN

Next Mr. Freeman made a study of reciprocating duplex pumps and developed the designs for the Underwriters' steam fire pump. Later on the details of rotary fire pumps were improved and standardized.

The automatic sprinkler was being developed in the latter '80's and early '90's, but there was no uniformity in the practice of spacing heads or arranging pipes. In 1892 Mr. Freeman conducted an elaborate series of experiments on sprinklers to determine the effective area of water distribution, and also a series of tests on the flow of water through pipes. Much valuable information was obtained on the friction loss in all sizes of pipes from $^3/_4$ in. to 8 in., together with the fittings, which was used as the basis for the present regulations for sprinkler spacing and supplies.

About the same time Mr. Freeman became interested in the improvement of construction of mills and worked out many of the details of the so-called New England mill-construction type of buildings. The construction of storehouses was studied and many typical structures were devised and adopted by the members. Some of the earliest buildings with the saw-tooth type of roof construction were designed in accordance with his views.

The first comprehensive set of electrical rules was compiled under Mr. Freeman's direction.

Mr. Freeman was placed in charge of the inspection department of the companies in January, 1890, and reorganized it along more efficient lines. He arranged for regular inspection service, systematic surveys and plans of risks, a laboratory for testing and research, and a division for adjustments and appraisals, as well as experts in special lines, such as mill construction, electricity, and chemistry. Tests were made on meters to determine the friction loss and a full flow meter designed that would be fairly accurate yet was free from the danger of becoming obstructed. Dry-pipe valves were developed in order to have automatic protective equipments in cold rooms. "Germ-proof" check valves were designed to prevent possibly polluted water being forced back into public mains.

In the early days of his connection with the inspection department there were a number of losses due to closed valves. All valves were underground; some turned to the right to open, others turned to the left, and there was no certainty as to the condition of the valve unless water was flowing. Mr. Freeman devised the so-called indicator post valve with its extension stem and target above ground to show clearly the condition of the valve. He was also responsible for the development the outside screw-and-yoke type of valve for inside piping.

All of the various features of the construction and protection of factories were studied intensively under Mr. Freeman's direction, and many safeguards were devised which proved a benefit not only to the manufacturers but also to the merchants and the public in general.

W. M. MCFARLAND, *Chairman*,
Committee on Awards and Prizes.

[1] Appendices to the Rules: In no case will this award be made until the invention or improvement shall have been fully described in MECHANICAL ENGINEERING for a period of thirty days.

A.S.M.E. Council Urges Prompt Trial of Indicted Engineers

AT a meeting of the Council of the A.S.M.E. held in Montreal, May 29, 1923, the following resolutions were passed in reference to the Department of Justice cantonment indictments and suits.

WHEREAS, the Council of The American Society of Mechanical Engineers, having an organization of approximately 16,000 members, has been shocked by the indictment of certain of the prominent members of the engineering profession on charges of conspiracy amounting to treason in time of war, and by the bringing of civil suits for many millions of dollars against some of the most reputable contracting organizations in the country, some members of which are honored members of the engineering profession, for alleged fraud in connection with the construction of the cantonments; and

WHEREAS, in the documents under which these actions were brought, it is stated that the findings of the Presidents of three National Engineering Societies and other members of the Committee appointed to pass on the contract adopted by the Emergency Construction Committee were not the result of investigation but of unconsidered substantiation of the actions of the Emergency Construction Committee; and

WHEREAS, in the text of the documents those indicted and those sued are repeatedly accused of delaying the Government's program of construction for the cantonments, while those who are well informed know that this work was done at an extraordinary speed; and

WHEREAS, wide publicity was given to those suits at the time they were brought, but now the officials of the Department of Justice are showing a disposition to delay in trying those actions; and

WHEREAS, the Constitution of the United States guarantees to every accused citizen a speedy trial.

Therefore, be it Resolved: That the Department of Justice, having challenged the integrity and honesty of purpose of prominent men in the engineering profession through these actions, the Council of this Society urge the prompt trial of these charges. Also that it is the duty of every member of this Society to demand through every avenue available that these charges be promptly tried.

And be it Further Resolved: That copies of this resolution be sent to the President of the United States and the Attorney General, and be printed in MECHANICAL ENGINEERING.

The actions in question were begun in November, December, and January. Indictments were secured against seven former Government administrative officials and agents as follows: Benedict Crowell, Cleveland, former Assistant Secretary of War; W. A. Starrett, New York, contractor; M. C. Tuttle, Boston, contractor; C. W. Lundoff, Cleveland, contractor; C. Foster, Major in the Engineer Corps during the war, engineer in construction business; J. H. MacGibbons, Chicago, representative of a bonding company; and J. A. Means, New York, contractor. These men, members of the Emergency Construction Committee during the war, are accused of frauds in contracts for erection of cantonments, port terminals, warehouses, and fortifications.

The firms against which suits on similar charges, totalling fifteen million dollars, were brought include many members of the engineering profession.

Similar resolutions were adopted by the Associated General Contractors of America, at their Annual Meeting, Feb. 2, 1923.

Objects and Methods of the F.A.E.S. Committee on Coal Storage

THE FEDERATED AMERICAN ENGINEERING SOCIETIES has undertaken the study of coal storage for certain distinct reasons. The main one is that in the extended and somewhat involved discussion on this subject, participated in by several Federal bureaus and Congressional committees, there has arisen a real need for an unbiased report on what is being done and what might be done in the storage of coal for the nation at large. The engineer is by nature the man who can meet this test. What we desire to secure are the facts pertaining to this important question.

In fulfilling this object the first step is to assemble the results of all investigations which have been made. A great deal of scientific work has been done, the reports on which are scattered through public reports. These are to be compiled so that the final report will contain authoritative statements as to the conditions under which coal may be stored. It is believed that such a compilation will serve a distinct purpose. We are fortunate in having on the committee Mr. S. W. Parr, professor of industrial chemistry at the University of Illinois, one of the leading authorities on the chemistry of coal.

One of the immediate functions of this committee is that of securing information for the use of the United States Coal Commission. This means that this work is closely associated with that of the Coal Commission and the Federal fuel distributor in the Department of Commerce. Information will be made available showing the coal requirements of various important industrial districts and of important groups of industries.

Closely associated with the phase of the work last mentioned, is the need expressed by the War Department for information on the fuel requirements of various manufacturing concerns which might supply munitions in case of war emergency. While this is incidental to the main work of the committee, it is expected that information of value may be derived.

It is evident to all that the leading motive prompting the investigation is that of determining the conditions under which storage might affect production in bituminous coal mining fields. For several reasons it is desirable that coal mining operations be stimulated in the Spring and Summer in order to cut down the intensity of demand upon the mines during the Fall and Winter season. The bituminous mines are operating only part time during warm weather, while the rush of orders at other seasons of the year necessitate the maintenance of a force of miners and of producing equipment capable of a peak-load production higher than is necessary. A primary object is to reduce this peak load, and create a more uniform demand throughout the year.

While it is not within the scope of the engineering investigation to initiate the measures to bring about a change, it is distinctly the purpose to furnish the information on which such efforts might be initiated by the proper parties.

It might be said that, in addition to the analysis of conditions under which coal may be stored, one of the most important portions of the work will be to determine the actual cost of storage. In order that industries and Governmental agencies may decide whether storage may be undertaken, it is necessary to know just what it costs.

It is not to be expected, of course, that coal will be stored unless users realize a financial advantage from doing it. To secure this information the combined experiences of all of the concerns which have stored coal are being sought. When such data have been accumulated, and conditions at the more notable coal storage plants have been studied, it is intended to develop typical statements as to the possibilities and suggestions for the best methods which might be employed in storage.

These in brief are the primary objects and methods which will guide the coal storage committee in its work. The support and coöperation of all engineers in the consummation of this work are most earnestly requested.

P. F. WALKER.[1]

[1] Dean of Engineering, University of Kansas; Field Executive, F.A.E.S. Committee on Coal Storage.

Southern Appalachian Power Conference Holds Annual Meeting

THE second meeting of the Southern Appalachian Water Power Conference, established in June, 1922, was held at Asheville, N. C., June 25–27, 1923. The adoption of the report of the Executive Committee on the future policy, work, and influence of the Conference was the outstanding event of the meeting. This report proposes the establishment of three bureaus to assist in carrying out the policy of the Conference, which is the "wise and beneficial development, utilization, and conservation of the great water powers, forests, and other natural resources of the Southern Appalachian states."

The bureaus which the plan contemplates are (1) a Bureau of Public Relations, "to effect a more harmonious relation between public utilities, public service regulatory bodies, and the general public;" (2) a Bureau of Industrial Development to "collect, collate, and disseminate accurate and reliable information regarding power and other factors in industrial expansion;" and (3) a Bureau of Research to "compile and digest technical data of value to hydraulic engineers, industrial chemists, and other professional men having to do with the utilization of the Natural resources of the South."

Resolutions were adopted urging the restoration of the original Congressional appropriation of $2,000,000 for the purchase of forest lands on the headwaters of streams, the increase of the appropriation for coöperation with the states in forest-fire prevention to at least $1,000,000 a year, and the extension of river navigation wherever possible by the building of navigation-power dams. J. H. Wills, Atlanta, Ga., was elected president for the coming year, and the name of the organization was changed to the Southern Appalachian Power Conference. Interesting papers on power development methods used in surveying a large river system, possible use of large crude-oil engine units for stand-by power plants, etc., were presented.

Federal Power Commission Reports on Distribution of License Fees for 1922

THE FEDERAL POWER COMMISSION announces that it has complete collection of the fees due the Government from operations during 1922 under licenses for water-power development, and has deposited the total of $29,519.23 in the Treasury for distribution and credit as follows:

To the general fund of the Treasury	$12,825.21
To the indefinite appropriation (under administration of the War Department): Maintenance and operation of dams and other improvements of navigable waters	12,386.41
To the Reclamation Fund	1,755.21
To Indian funds	1,236.00
To Payments to states under the Federal water-power act, special funds	1,316.40
Total	$29,519.23

The Commission's receipts from this source during 1921 were $8,963.57. For 1923 they are expected to be $60,000. Its operating expenses, exclusive of salaries of assigned personnel, for the years 1921–1923 will not exceed $107,000, as now estimated. It is expected that within five years from the date of approval of the act (June 10, 1920) the charges collected will have offset all expenses, direct and indirect, in its administration.

Coffin Foundation Makes First Award

THE Southern California Edison Company, Los Angeles, has been selected from eighteen electric light and power companies which submitted records for consideration, as the one which has made the most distinguished contribution to the development of electric light and power for the convenience of the public and the benefit of the industry during the past year. The award, consisting of a gold medal and one thousand dollars, was presented to John B. Miller, president of the company, by Frank W. Smith, chairman of the prize committee of the Charles A. Coffin Foundation, during a meeting of the public policy committee of the National Electric Light Association on June 7, 1923.

Captain Robert Woolston Hunt

LESS than a month after receiving the high honor of the Washington Award "for his pioneer work in the development of the steel industry in the United States and for a life devoted to the advancement of the engineering profession," Captain Robert Woolston Hunt died at his home in Chicago on July 11, 1923.

Captain Hunt was born December 9, 1838, in Fallsington, Bucks County, Pa. His father, Dr. Robert A. Hunt, died when he was only sixteen years of age, and in 1857 the family moved to Pottsville, Pa., where Robert spent several years in the iron rolling mill of John Burnish & Co., acquiring a practical knowledge of puddling, heating, rolling, and other details of the iron-rail business. Later he took a course in analytical inorganic chemistry in a laboratory in Philadelphia, upon the completion of which he entered the employ of the Cambria Iron Works at Johnstown, Pa. The first analytical laboratory maintained in America by an iron and steel company was established under his direction in 1860 for the Cambria Works.

In the Spring of 1861, as night foreman of the Elmira rolling mill, Elmira, N. Y., Captain Hunt assisted in the organization of the working force and the starting of that industry. In the Fall of that year he entered the U. S. Army, and was stationed at Harrisburg. In 1862 he was put in command of Camp Curtin, at Harrisburg, with the rank of captain. The next year he served the state of Pennsylvania as mustering officer, and in 1864 assisted in recruiting Lambert's Independent Mounted Company of Pennsylvania Volunteers.

After the close of the war Captain Hunt returned to the employ of the Cambria Iron Company and was sent to the experimental Bessemer works at Wyandotte, Mich., owned jointly by the Cambria Company and the Kelly Pneumatic Process Company. The Kelly plant had in 1864 produced from pig iron remelted in a reverberatory the first pneumatic steel made in America, and Captain Hunt, while in charge of the works there, introduced the use of a cupola for remelting. In 1866 he returned to Johnstown, where he had charge of the hammering and rolling of steel and superintended the rolling of the first steel rails made in America on a commercial order, produced for the Pennsylvania Railroad from ingots cast by the Pennsylvania Steel Company's plant at Steelton, Pa. The behavior of these ingots led to the successful introduction of blooming by rolling instead of hammering.

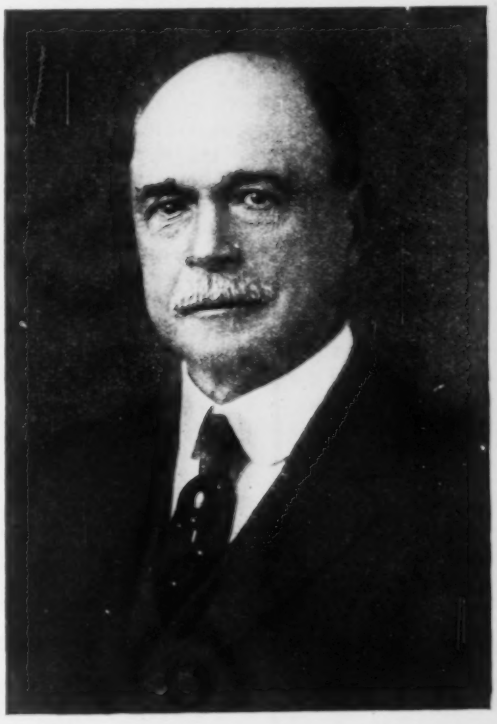
The Bessemer plant of the Cambria Company, finished in 1871, was designed and built by John Fritz, chief engineer of the company, with the assistance of A. L. Holley and Captain Hunt. It embodies several new features, such as the continuous running of the cupola, and a new system of bottom casting. Captain Hunt was in charge of these works from the time of their completion until August, 1873, when he resigned to become superintendent of the Bessemer Works of John A. Griswold & Co., at Troy, N. Y. These works were enlarged and remodelled, and the name of the company changed, first to the Albany & Rensselaer Steel & Iron Co., and later to the Troy Iron & Steel Co. Captain Hunt remained in charge until 1888, and during these years he supervised the erection of three large blast furnaces, and made many grades of Bessemer steel not previously produced in America, notably soft steel for drop forgings. He was also a pioneer in the production of steel for gun barrels, carriage axles, drills, and springs. He took out several patents on steel and iron metallurgical processes and machinery, and the Hunt-Jones-Suppes rail-mill feed tables developed during this period, were used later under licenses by the majority of the rail mills in the United States.

In 1888 Captain Hunt resigned his position at Troy and went to Chicago, where he established the firm of Robert W. Hunt & Co., consulting engineers, an organization which now has offices and laboratories in many cities.

Scientific and engineering ability and strict integrity characterize the business enterprise which bears his name, and one of the sources of great satisfaction to Captain Hunt as he approached the end of his long life was that he had been able to place this organization upon a basis which will enable those who were his supporters and assistants to continue the work indefinitely.

The Washington Award is one of many tributes to the work of Captain Hunt. In 1912 he was awarded the John Fritz Medal, "for his contributions to the early development of the Bessemer Process," and in 1920 the Hunt Medal "for meritorius contributions to the art of making steel." He was for many years a trustee of the Rensselaer Polytechnic Institute, which in 1916 conferred upon him the honorary degree of Doctor of Engineering. While residing in Troy he was elected for four successive terms Commander of John A. Griswold Post, No. 338, G.A.R.

In the development of the engineering societies he has played an important part. He was president of the A.I.M.E. in 1883 and again in 1906; president of the A.S.M.E. in 1891; president of the Western Society of Engineers in 1893; president of the American Society for Testing Materials in 1912; and American Vice-President of the International Association for Testing Materials in 1914. He belonged also to the A.S.C.E., the Institution of Civil Engineers, the Institution of Mechanical Engineers, the Canadian Society of Civil Engineers, and the British Iron and Steel Institute. Captain Hunt was secretary of the Committee on Standard Rail



ROBERT W. HUNT

Sections appointed by the A.S.C.E., the final report of which was made in 1893, and of the Special Committee on Rail Sections, appointed in 1892, which reported finally in 1910. The sections recommended by this committee were generally adopted by the railways of America.

His services as an officer and on committees of these societies, as well as his contributions to their proceedings, and lectures before educational and scientific bodies, have been of inestimable value. He was an honorary member of the A.I.M.E., A.S.M.E., American Society for Testing Materials, and Western Society of Engineers.

Captain Hunt was a member also of many athletic and social clubs, and in them, as in all his contacts with his fellow men, had many friends. He placed high value upon these friendships, saying in one of his last public addresses, at the time of the presentation of the Washington Award, "If ever a man has lived who has been blest with the friendship of his fellowmen, it has been myself. Never in my long life have I experienced a moment that, for any reason, I could say I was without friends."

Engineering and Industrial Standardization

Secretary Hoover Advocates Uniform Specifications in Making Government Purchases[1]

I HAVE, with the approval of the state governors, called this conference upon "Specifications" for purchase of supplies by our national and local governments and public institutions. This bald statement of purpose might appear to the uninitiated as a question of little importance. As a matter of fact in its ramifications it means hundreds of millions of dollars to the taxpayer and much more to the private consumer and to the manufacturer. It is an important link in the whole great chain of elimination of waste in our industrial and commercial fabric.

Specifications are the formulated, definite, and complete statements of what the buyer requires of the seller. They must be an accurate representation of the chemical and physical characteristics of the article or material required—and they must be adapted to the best practice of production and distribution.

The first step in formulation, therefore, is the determination of these chemical or physical characteristics and this brings us to the entire field of standardization of commodities; that is, the determination of qualities, of grades, of purposes and performance. Scientific standards, of course, are of old origin; Emerson defined a standard as "That which is established by investigation or authority to be reasonably attainable maximum of desirability." Their formulation requires the services of chemists, physicists, engineers and something that has been often neglected, the added experience of the manufacturer, the producer and the user. There is no economy to be obtained.

Standardization insures quantity production and hence lower production cost; smaller investment of capital and, therefore, the release of capital for other industrial purposes and developments.

Specifications and standards are a changing requirement, for there is steady progress in the arts and invention; and requirements, therefore, must be subject to periodic review.

Being aware from war experience of the great faultiness of specifications used in Federal purchases, I undertook early in this administration that the Department of Commerce should review these specifications. Subsequently, with the establishment of the Bureau of the Budget, an interdepartmental committee was created for the better development of this work, at the suggestion and under the leadership of the Department of Commerce. Each of the specifications are being taken in hand and the chemical and physical requirements of each article in question are being exhaustively examined in the laboratories of the Bureau of Standards. Beyond this, however, after we have arrived at definite scientific background, the manufacturers in each branch are being brought into consultation to make sure that the industrial and commercial setting of a given specification is right from the point of view of the practical producer. By so doing we have found that a great many specifications for large purchases were loosely drawn, so that the various manufacturers making bids were offering really entirely different grades and qualities, and that, therefore, a mere monetary difference in price did not denominate relative merits because of the lack of precision in specifications. Moreover, we have found that other specifications were drawn purely from the point of view of the personal vision of some official, and as drawn required special manufacture, when well-known articles, already standardized, necessarily bid a price that would warrant special production and consequent interruption of ordinary business.

Now, all these reorganized specifications are enabling the Government to buy on much better terms and at savings to the taxpayer. This conference, therefore, has been called with the thought that the Federal government—with its great laboratories and with its large experience in the standardization of purchases, in the formulation of the sequent specifications, and in the development of methods of testing—could be of service to the purchasing agencies of states, municipalities, and public institutions.

Especially in research work and in the elaboration of methods of testing can the Federal Government and its laboratories be of value to purchasing agencies and industry generally. Research is a necessary component of standardization and the formulation of specifications.

This whole question is proving to be of vastly greater importance than even the savings on public purchases might imply. Industry itself has long struggled to establish standards and to secure their acceptance as a method of simplifying the process of manufacture and raising the ethics of production. Furthermore, where standards and specifications are successfully developed by the Government through the methods we have undertaken, they are receiving wide voluntary acceptance and adoption among public consumers. I need only to point to the case of cement, where practically the whole output of the country is now produced and distributed on the basis of Federal standards and tests—it being sufficient in the purchase of cement simply to state "Federal specifications."

Now I do not wish to give the impression that specifications must be newly discovered in order to be proper and advantageous. Our engineering and professional societies, especially the American Society for Testing Materials and the American Engineering Standards Committee, have long been engaged in the preparation and standardization of certain types of specifications; our university laboratories, our great industrial laboratories, the engineers in our large industries, have for years been developing specifications in a great multitude of materials. Many of these specifications have lain dormant for lack of use by the consumer in his demands upon the producer. Others of them have not had applied to them the test of commercial experience in production. It is our purpose to adopt whatever is good, to put it under review as to its practical application, and, as we have done hitherto, to call in the expert manufacturer and producer for his coöperation and advice.

There are a great many directions in which this problem of waste elimination and increased efficiency extends. It extends even into such problems as an ample transportation system. But a very large opportunity for the elimination of waste and the increase of efficiency lies in the field that I have been describing to you today. It is a field of much larger dimensions, in hundreds of millions and billions of dollars, than will be believed by any except men of professional experience; it is proposed to seek your coöperation in this field, not only for the purpose of decreasing the cost of supplies in public purchases, but also in the influence that the consumer can exert in coöperation with the producer to minimize the costs of production and distribution.

I could formulate these propositions into terms of reducing the cost of living, or I could formulate them into terms of raising standards of living. Whichever economic phrase we may adopt, they both lead to the same end—and that is the increased comfort and happiness of our people.

[1] Abstract of address delivered at a Conference of State Purchasing Agents, Washington, D. C., May 26, 1923.

Items of Interest in the Standardization Field

Telephone Group Organized. At a recent meeting of the U. S. Independent Telephone Association it voted to take membership in the A.E.S.C. with one representative. This Association with the American Telephone and Telegraph Company will therefore constitute the Telephone Group.

N. A. of P. A. Discuss Standardization. Dr. P. G. Agnew, Secretrary of the A.E.S.C., told the National Association of Purchasing Agents at their annual convention in Cleveland that the purchasing agents of state and municipal government departments are spending approximately $700,000,000 yearly for supplies, materials of construction, and other products. During his address he pointed out how much of this amount becomes a direct waste, then told of what his committee is doing to eliminate a part of this waste.

Methods of Analysis of Paint Pigments. Many of the controversies growing out of differences in methods of analyzing paint pigments should be eliminated by the universal use of the method for routine analysis of white pigments, yellow, orange, red and brown pigments, containing iron and manganese, and of dry red lead, which have just been approved as Tentative American Standards by the

American Engineering Standards Committee. The approved methods provide a basis for the analysis of paint pigments, which has always been a complicated and often a controversial problem. The specifications for these methods of analysis were presented for A.E.S.C. approval by the American Society for Testing Materials under whose auspices the specifications were developed, and the A.S.T.M. has been designated as sponsor for these specifications.

Sponsors for Road Material. The American Engineering Standards Committee announces the appointment of the following four organizations as joint sponsors for the specifications of the Method of Test for Penetration of Bituminous Materials submitted by the American Society for Testing Materials: American Association of State Highway Officials, American Society for Municipal Improvements, American Society for Testing Materials, and the Bureau of Public Roads. This is only one of several standardization projects involving road materials now before the American Engineering Standards Committee,, the others covering the toughness of rock, the distillation of bituminous materials, and specifications for concrete.

Paper-Stock Sheet Sizes. One of the net results of a broadly representative conference which met in Washington on June 19 was the final adoption by the manufacturers' and users' organizations of the following four standard sheet sizes for book-paper sizes:

25 in. by 38 in.	38 in. by 50 in.
26 in. by 29 in.	29 in. by 52 in.
32 in. by 44 in.	44 in. by 64 in.

35 in. by 45½ in.

This conference also agreed that the standard method of determining the basic weight or "substance" of a given sheet of paper shall be on a ream 25 in. by 40 in. of 500 or 1000 sheets for all paper.

Aeronautical Safety Code. The Sectional Committee which is developing the American Aeronautical Safety Code held a meeting recently at the Bureau of Standards. At this meeting the committee adopted reports of two sub-committees consisting of rules for balloons, airships, landing fields, seaplane stations, and airways. These parts of the code are now available for general use and application.

Steel Bridge Standards. The American Railway Engineering Association has submitted its General Specifications for Steel Railway Bridges and its Specifications for Movable Railway Bridges for approval by the American Engineering Standards Committee as American Standards.

Protection Against Lightning. The Sectional Committee on Protection Against Lightning organized jointly by the Bureau of Standards and the American Institute of Electrical Engineers, has been active in the preparation of a tentative standard for the protection of buildings and other property against lightning. A joint meeting of Group I of the Sectional Committee in charge of the work, together with the National Fire Protection Association's Committee on the same subject, was recently held in Chicago. Copies of the tentative standard have also been circulated generally for comment by those interested, and the final draft is now being worked out.

Roller Chains and Sprockets

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las are special for cutters covering the standard ranges of teeth:

$$W = 1.24D \cos Yab; \quad V = 1.24D \sin Yab$$

$$yz = D\left[1.24 \sin\left(17° + \frac{148}{T} - Yab\right) - 0.8 \sin\left(18° - \frac{28}{T}\right)\right]$$

$$F = D\left[0.8 \cos\left(18° - \frac{28}{T}\right) + 1.24 \cos\left(17° + \frac{148}{T} - Yab\right) - 1.3\right]$$

$$cb = c'b' = \sqrt{(E + F)^2 + yz^2}$$

Width of cutter = Q = 1.02 sin (180°/n) × outside diameter of the smallest sprocket in the range to be cut, to the next higher 32nd inch.

Thermal Conductivity of Liquids

(Continued from page 482)

B.t.u. per hr. This rate is taken at the surface B or 0.81 in. below the heater. Hence

$$K = \frac{300 \times 4.13}{(180 - 80) \times 1 \times 3} = 4.13.$$

Correcting for losses as explained above, K becomes 4.13 × 1.16 = 4.79 B.t.u. per sq. ft. per degree difference in temperature per inch per hour.

The computed values of K are given in Table 2. In this table the 1.16 correction has not been applied. It should be noted that, partly owing to reasons of time, computed results are tabulated only for temperature gradients $(T_1 - T_2)/L$ having values between 17 to 24, although the gradient values varied between 4 and 43.

For purposes of comparison, the available data upon the conductivity of water and their variation with temperatures were collected and plotted as in Fig. 7. This material was in c.g.s. units and was converted into the English system by multiplying by 2.90 × 10^2. The names of the authorities are given, and it can be seen that there is a wide variation in the results. The curve drawn through these points was made upon the popular assumption that K varies directly as the temperature.

As can be seen from Fig. 7, the conductivity as determined from these tests falls below the assumed curve based on results of previous investigations. It was unfortunate that the average temperature was not low enough to give a better chance of checking. The only close check is for 110–115 deg. Here Lees obtained 3.5 while the value in the author's test was 3.65. Lees' data contradict those of other authorities, K decreasing with an increase in temperature in his results.

Discussion of Results

In carrying out the work there were a number of sources of error that were not eliminated. For instance, the results should be multiplied by 1.16 to correct for heat losses, assuming the condensate method of measuring the heat input to be correct. There is uncertainty as to just what takes place between the heater and the liquid surface. The thermocouples indicate a condition of turbulence, which would lead to the belief that the surface contact was not good. If this were so, considerable heat would be lost in penetrating the pocketed air space. With good surface contact, another factor might enter into the problem. With the top water layers at higher temperatures than the bottom ones, imperfect or unequal expansion would take place. This would tend to increase the pressure within the liquid and might be a source of convection currents. Also, with continuous heating, the temperature of the bottom layers could not be brought to the temperature of the top layers, and this is an indication that heat was leaving through the bottom of the vat or that the water-logged vat became finally a better conductor than the vat.

Notwithstanding these sources of error, a number of tentative conclusions may be drawn. On the whole, the conductivity seems to vary inversely with the temperature, and in Fig. 8, with curves drawn the various points, two curious things are noticeable, the conductivities for the same temperature gradient $(T_1 - T_2)/L$ lie upon a straight line, and the conductivities themselves vary greatly for the same temperatures. Thus at 130 deg. fahr., K is found to be 1.5 and also 4.15. Experimental error may be responsible for this peculiar phenomenon, but in the opinion of the author K depends upon the temperature gradient and not upon the average temperature.

The foregoing conclusions are purely tentative, and the author believes that the subject must be investigated more carefully and with better-constructed apparatus. A container of bakelite might be used. Some means should be available for keeping $(T_1 - T_2)/L$ constant. The source of heat should be of greater intensity than that afforded by steam, so that a greater quantity could be supplied in a given time. Control of the surface contact would be an advantage. Means of measuring the heat passing each section, if one could be devised, would settle the great difficulty of obtaining the flow rate. Lastly, a more sensitive and more accurate means of measuring temperatures would be desirable.